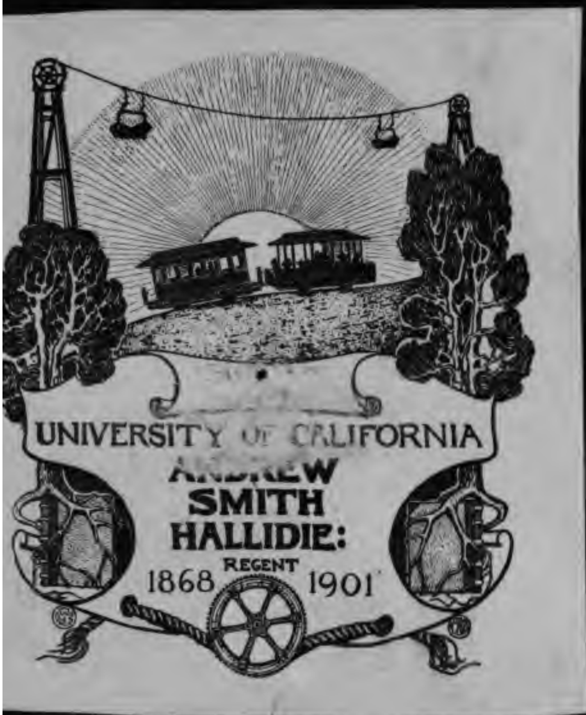
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THE STEAM TURBINE

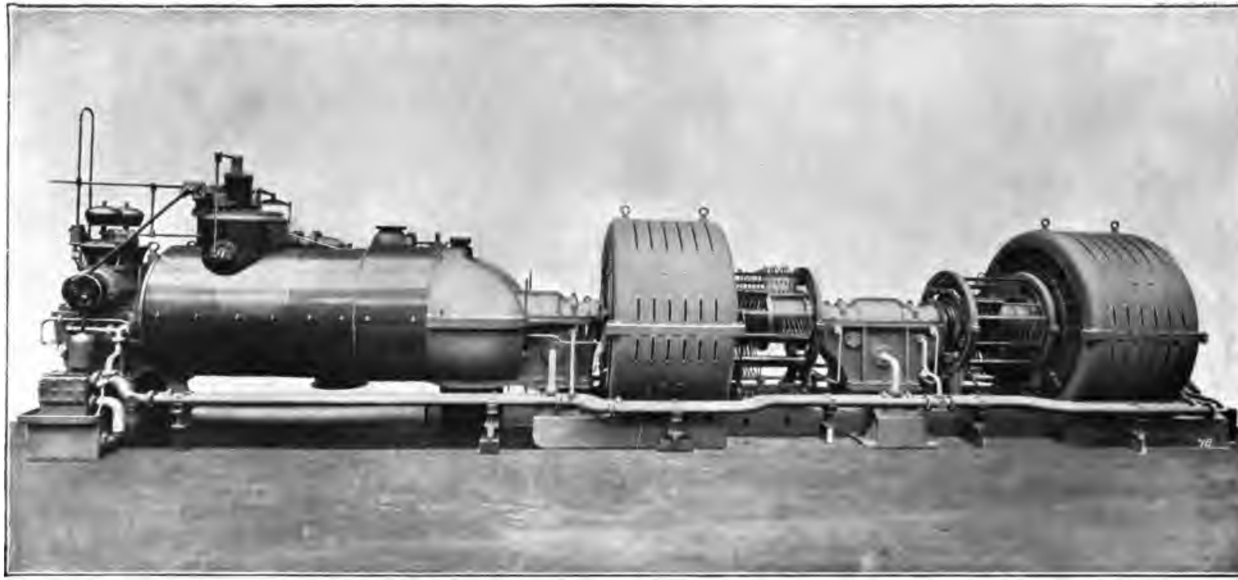

PLATE I.—1800-KILOWATT PARSONS TURBO-DYNAMO AT MANCHESTER CORPORATION (DICKINSON STREET) ELECTRICITY WORKS.

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BY

[illegible] M. NEILSON

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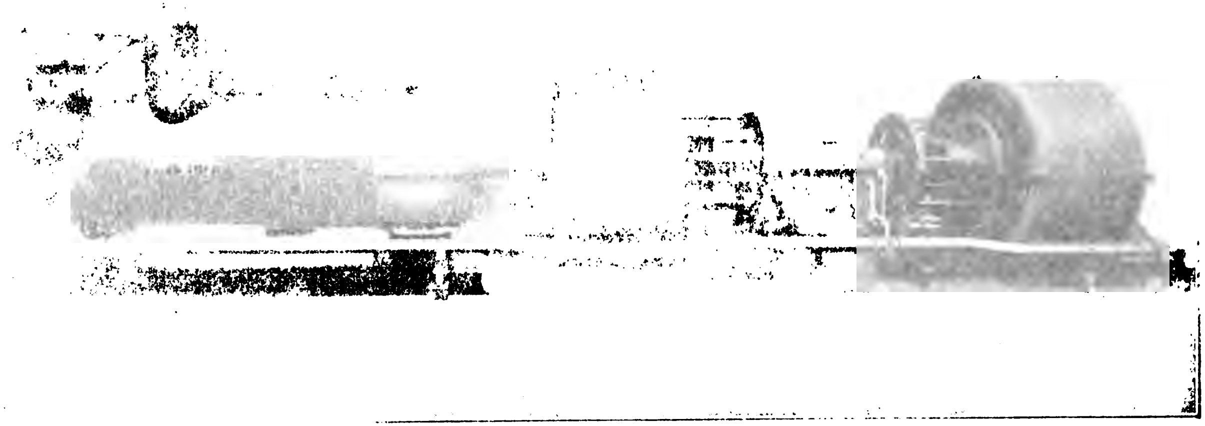

Dynamo at Manchester Corporation (Dickinson Street) Electricity Works.

THE
STEAM TURBINE

BY

ROBERT M. NEILSON

WHITWORTH EXHIBITIONER; ASSOCIATE MEMBER OF THE INSTITUTION OF MECHANICAL ENGINEERS; MEMBER OF THE MANCHESTER ASSOCIATION OF ENGINEERS; FELLOW OF THE CHARTERED INSTITUTE OF PATENT AGENTS; LECTURER ON STEAM AND THE STEAM ENGINE AT THE HEGINBOTTOM TECHNICAL SCHOOL, ASHTON-UNDER LYNE

WITH NUMEROUS ILLUSTRATIONS

THIRD EDITION. REVISED
RE-ISSUE, WITH ADDITIONS

LONGMANS, GREEN, AND CO.
39 PATERNOSTER ROW, LONDON
NEW YORK AND BOMBAY
1905

PREFACE TO THE FIRST EDITION

That the steam turbine is likely to be extensively used in the future is admitted by most engineers; but, although a good deal has lately been written about this type of engine, this literature has mostly consisted of descriptions of the principal features only, or of accounts of the results of tests.

The author has endeavoured in this book to describe, not only the principal parts of the leading types of steam turbine, but also the small details which, in the case of this motor, have such a preponderating influence in determining success or failure. The theory of the action of the steam turbine is also treated of, and the subject is likewise dealt with historically.

Comparisons have necessarily been made with the hydraulic turbine and with the reciprocating engine; but, with a view to extending the usefulness of the book, the author has assumed on the part of the reader no prior knowledge of the hydraulic turbine, and only an elementary knowledge of the reciprocating engine and of the laws of thermo-dynamics.

With a like object in view the author has tried to make the mathematical reasoning as simple as possible.

As entropy-temperature diagrams are not yet widely understood, a chapter on this subject has been given; but the matter has been treated as briefly as possible.

The results of tests of steam turbines given throughout the book have been carefully selected with a view to obtaining the strictest accuracy.

The author takes this opportunity of thanking the various individuals and firms who have given him information and assistance, and of expressing his indebtedness to Messrs. C. A. Parsons and Co., Newcastle-on-Tyne, and the Société de Laval of France for the loan of several blocks.

R. M. N.

30, Cross Street, Manchester,
June, 1902.

PREFACE TO THE SECOND EDITION

The chapters descriptive of the Parsons and De Laval steam turbines have been considerably enlarged, and I hope that the many deficiencies of the first edition have in great measure been remedied.

The chapter on marine propulsion has been greatly extended, so as to give a record of the developments of the subject during the past year; and several illustrations of propelling machinery have been added. The chapter on vanes and velocities has also been extended.

A new chapter has been added in which the Westinghouse-Parsons, the Stumpf, the Schulz, the Curtis and the Seger steam turbines are described; and another new chapter deals with the question of the saving of space obtained by employing high speeds.

The original appendix has been brought up to date, and a second appendix added, which it is hoped will prove, if not of great value, at least of some convenience to readers.

I take this opportunity of expressing my thanks to the various individuals and firms who have kindly supplied me with particulars for this edition. Among these I should like to mention Messrs. C. A. Parsons and Co., The Parsons Marine Steam Turbine Co., Ltd., Messrs. Greenwood and Batley, Ltd.,

The American De Laval Steam Turbine Co, The Westinghouse Companies' Publishing Department, Messrs. Wm. Denny and Bros., Messrs. Yarrow and Co., Ltd., and Mr. A. J. Tonge, of the Hulton Collieries, who have supplied me with some excellent photographs.

R. M. N.

30, Cross Street, Manchester,
August, 1903.

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THE STEAM TURBINE

CHAPTER I.

GENERAL REMARKS ON TURBINES.

A TURBINE is a machine in which a rotary motion is obtained by the gradual change of momentum of a fluid.

Fig. 1 shows a turbine diagrammatically. The partitions B between the passages A are called vanes, or blades, or buckets. Now, it is obvious that, if a fluid enters the space between two vanes in the direction shown by the arrow 1, and leaves in the direction shown by the arrow 2, the component of its velocity perpendicular to the radius will gradually change in its passage. The component might not change during the whole of the passage of the fluid owing to the vanes themselves having a velocity; but it will have a gradual change during at least some part of this passage. The fluid, therefore, has its momentum gradually changed, and it is this change of momentum which causes the vanes to rotate. The turbine wheel in the figure would rotate in the direction of the arrow 3. The action of the fluid on the

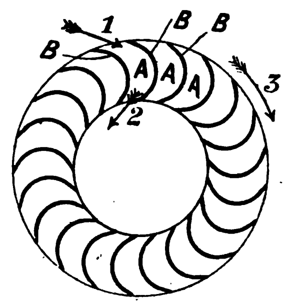


FIG. 1.—Diagrammatic Illustration of Turbine.

turbine will be discussed more fully later on; it is only desired at present to give a general idea of a turbine.

Turbines may be classified in several ways. Firstly, they may be classified according to the actuating fluid. The fluids most commonly used are water and steam, and the turbines actuated thereby are called respectively hydraulic turbines and steam turbines.

Turbines may be classified according to the direction of flow of the fluid into three classes: (1) In **radial-flow turbines** the fluid travels from the centre to the circumference of the wheel, or from the circumference to the centre. This class is subdivided into **outward-flow** and **inward-flow** turbines, according as the fluid passes from the centre to the circumference, or from the circumference towards the centre. (2) In **parallel-flow** or **axial-flow turbines** the direction of the flow of the fluid is parallel to the axis of the wheel, or in a spiral co-axial with the wheel. (3) In **mixed-flow turbines** the fluid flows both as in a radial-flow and as in a parallel-flow turbine.

Turbines are classified in other ways besides these; but as the other ways are not of importance, or do not hold good with steam turbines, we shall not refer to them.

Fig. 2 illustrates the principle of a **parallel-flow De Laval** steam turbine. The steam reaches the wheel by way of the divergent nozzles, where it expands and attains a great velocity. With this velocity it impinges on the vanes of the wheel, and causes the latter to rotate at a high speed. The wheel is enclosed loosely in a box or case, from which the steam escapes to the atmosphere or to a condenser. A section of one of the nozzles is shown at Fig. 2A drawn to an enlarged scale. In this figure the dotted line indicates the axis of rotation of the

wheel. The De Laval turbine will be more fully described later on.

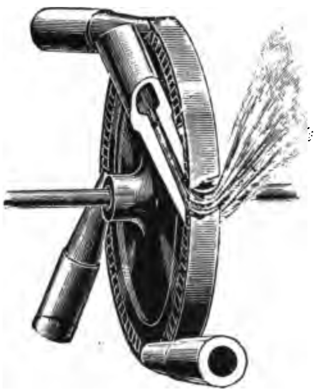


Fig. 2.—Action of Steam in De Laval Turbine.

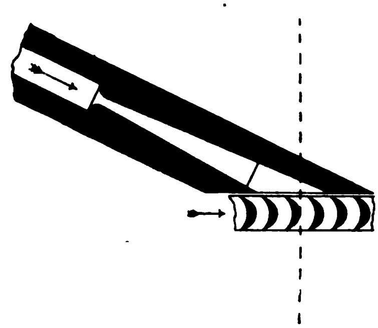


Fig. 2A.—Section of Nozzle of De Laval Steam Turbine.

Figs. 3, 4, 5, 6, 7, and 8 illustrate parts of a **Parsons parallel-flow** steam turbine. In this turbine the steam acts successively on a number of rings of blades. Part of one of these is shown in perspective view in Fig. 3, in elevation in

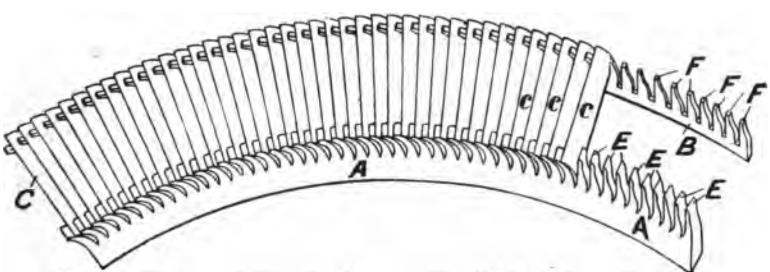


Fig. 3.—Blades and Shrouds of Parsons Parallel-flow Steam Turbine.

Fig. 4, and in plan in Fig. 5. Each ring of blades in this example is formed of blades, *c*, gripped in suitable recesses in shrouds, A and B. The rings thus formed are fixed alternately to the inside of the fixed cylindrical casing of the turbine, and to a revolving drum mounted inside the casing. Figs. 6

and 7 show parts of the casing and drum, the casing being lettered I and the drum H. Fig. 6 is a section taken through

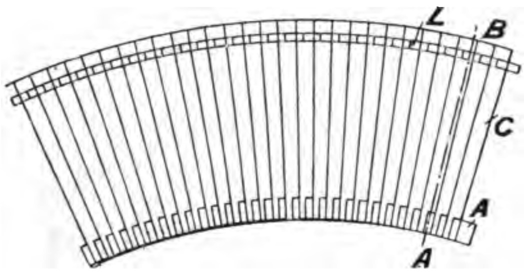


FIG. 4.

FIG. 5.

Blades and Shrouds of a Parsons Parallel-flow Steam Turbine.

the axis of the casing, while Fig. 7 is a cross-section on the line CD of Fig. 6. Power is obtained from the spindle G,

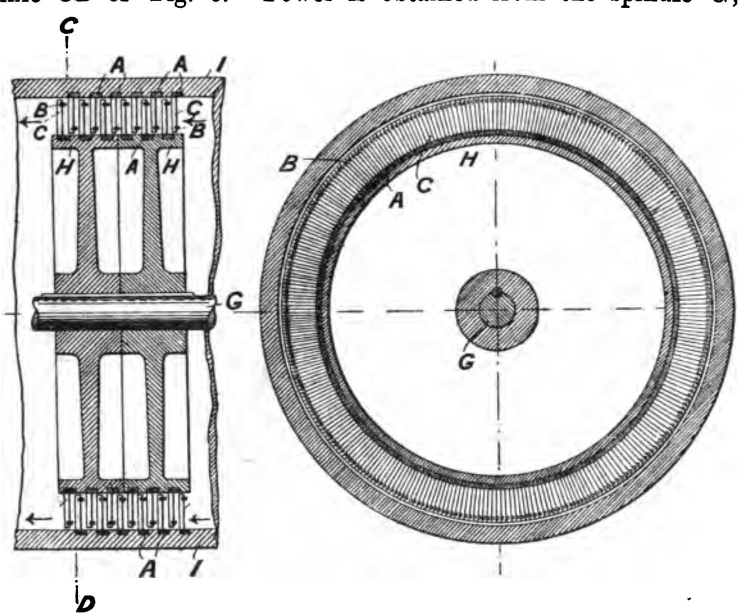


FIG. 6.—Partial Axial Section. FIG. 7.—Cross-section.
Parsons Parallel-flow Steam Turbine.

on which the drum H is keyed. It will be seen that the larger shroud A of each ring is secured to the casing or drum, while the smaller shroud B is free. The steam passing in the direction of the arrows in Fig. 6 acts on the moving blades so as to rotate them, and with them the drum and spindle. The fixed blades serve as guides to cause the steam after leaving one ring of moving blades to impinge in the right direction on the next ring of moving blades. The action of the steam on the blades can be clearly seen in Fig. 8, where the vertical arrows show the direction of motion of the moving blades and the horizontal arrows the direction of flow of the steam. This figure, which is drawn about full size, also shows the size, shape, and arrangement of the blades. These particulars, however, vary somewhat.

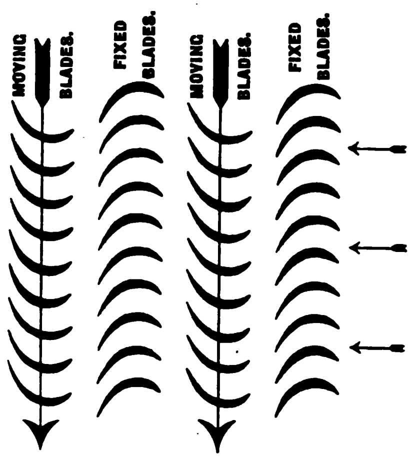


FIG. 8.—Action of Steam on the Blades of a Parsons Turbine.

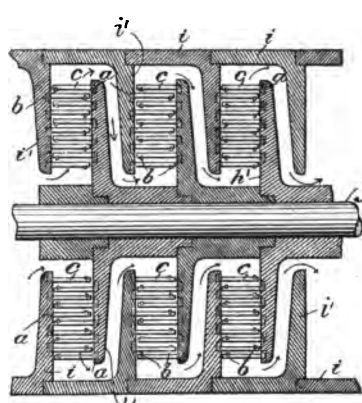


FIG. 9.—Parsons Radial-flow Steam Turbine, Partial Axial Section.

Fig. 9 is a partial axial section through a **Parsons radial-flow** turbine, and Fig. 10 illustrates a ring of blades for the same drawn to an enlarged scale. The blades *c*, both fixed and moving,

are held in shrouds, *a* and *b*, of a similar nature to the shrouds A and B of the parallel-flow turbine. The cylindrical casing *i* carries internal annular flanges, *i'*, to which are attached the larger shrouds *a* of the fixed rings of blades; while the similar shrouds of the moving rings of blades are supported on annular flanges, *h'*, carried by the spindle *g*. The smaller shrouds *b* of

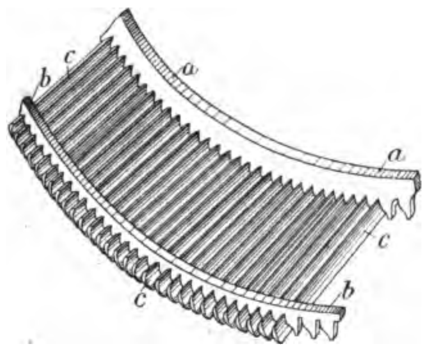


FIG. 10.—Blades and Shrouds of a Parsons Radial-flow Steam Turbine.

both fixed and moving rings are left free. The path of the steam is indicated by the arrows in Fig. 9, and it will be seen that the steam acts on the moving blades while flowing radially outwards in several stages.

The Parsons turbine in its several forms will be more fully described afterwards. The short description just made will, however, give a general idea of its nature.

CHAPTER II.

HISTORY OF THE STEAM TURBINE.

GOING back long before the days of Watt and Newcomen, we find a reaction steam-engine mentioned by the Egyptian philosopher **Hero** in his book on "Pneumatics," written in the second century B.C. This engine consisted of a hollow sphere rotating on two trunnions, through one of which it received steam from a generator situated below the sphere. The sphere was provided with two opposite projecting arms at right angles to the axis of the trunnions, the arms being furnished each with a nozzle at right angles to the arms and to the plane containing the arms and the trunnions. The nozzles were pointed in opposite directions, and the steam which escaped by them from the sphere caused the rotation of the latter about the trunnions.

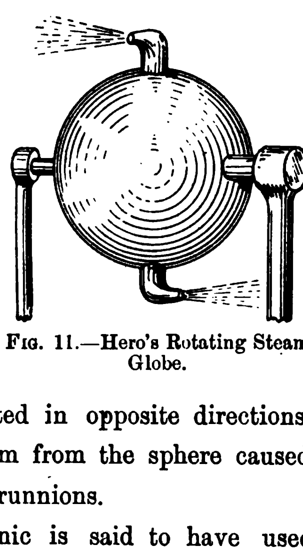


FIG. 11.—Hero's Rotating Steam Globe.

In A.D. 1577 a German mechanic is said to have used Hero's engine to rotate a broach in place of a turnspit.

In 1629 an Italian architect named **Branca** described a steam wheel or turbine in which a jet of steam was projected against a series of vanes on a rotating wheel.

In 1642 a Jesuit named **Kircher** used Branca's wheel, but with two jets of vapour acting on its circumference instead of only one.

In 1784 **Wolfgang de Kempelen** was granted a British patent for "Obtaining and transmitting motive power." The patentee thus describes his invention—

"When the machine acts by boiling water, or rather the vapour proceeding therefrom, a boiler is to be constructed (A, Fig. 12) furnished with a valve of security (B), the weight

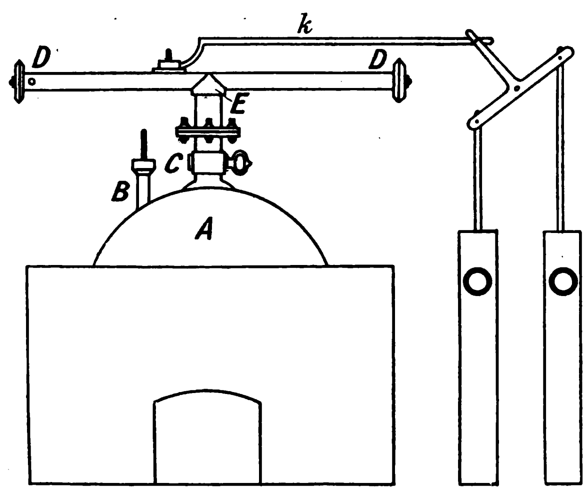


Fig. 12.—Wolfgang de Kempelen's Turbine.

of which is to be proportioned to the strength of the machine. At the upper extremity of the boiler is to be fixed a turn-cock (C), upon which the cylinder (DD) is to be screwed, the form of which cylinder appears in Fig. 13, where DD is a hollow cyclinder or tube, in the centre of which E is an aperture to contain the worm of the screw. FF is a tube of cast iron, having at the lower extremity a circular projection or plate, which, when this tube is pushed into the other

tube, GG, fills up the cavity therein marked (*aa*), so that the screw (*bb*) extends beyond the utmost length of the tube GG. Upon this screw the cylinder DD, with its nut, is to be fixed, and upon the plate of the tube GG of brass is to be screwed another plate (HH) of equal dimensions, so that the little plate, when it is in the cavity (*aa*), may be enclosed between

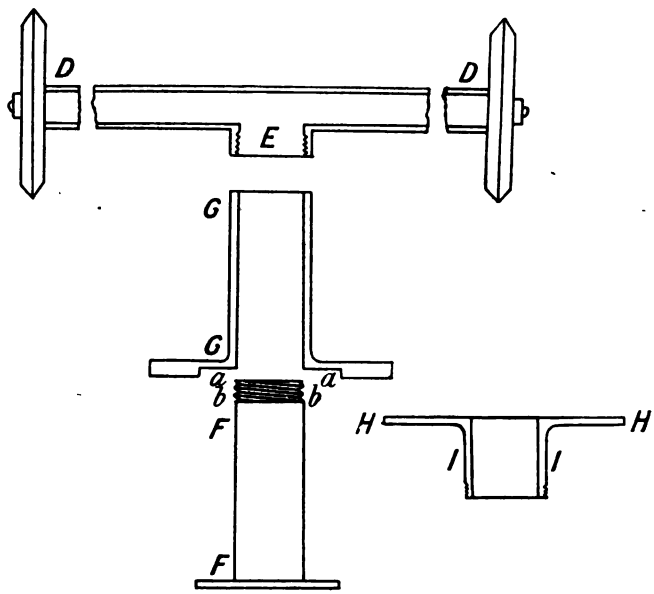


Fig. 13.—Details of Kempelen's Turbine.

two plates, and the tube FF left at liberty to turn round The plate HH has also a short tube (II) of an equal aperture with the tube GG, and at the end of it a screw is fixed, which surrounds the cock C, and is fastened within. Near each extremity of the cylinder DD, but on the contrary sides, is a small aperture, the size of which must be commensurate to the extent of the superfices of the boiling water, as, for instance, when the boiler measures within six feet in diameter,

requiring a valve of security weighing five pounds, the aperture near each end of the moving cylinder must be one inch in diameter. To put the machine in motion when the vapour of the boiling water is found strong enough to lift up the valve, the cock (C) is to be opened; the vapour instantly rushes through, and fills the cylinder DD, and finding a vent through the small apertures near its extremities on different sides, drives the cylinder round by reaction with exceeding great velocity. Having accomplished this first moving power which constitutes the principle of the machine, any kind of machine or engine may very easily be put into motion by it by means of a handle crown-wheel pinion, or other connection adapted to it, as is done with respect to a double pump by the excentric trunnion, *k*, Fig. 12."

The patentee then describes in his specification how his engine can be worked by water conveyed from a height, or by water acted on by steam pressure. The last-mentioned method is not illustrated, but the patentee states that two receivers of iron or copper must be provided between the boiler and the turning cylinder, and connected with both. The steam from the boiler is admitted alternately to the two receivers, and, pressing on the surface of the water, forces this into the turning cylinder, and rotates the latter by its reactive force when issuing from the apertures at its ends. The water is returned to the receivers.

In the same year **Watt** was granted letters patent for certain improvements relating to steam-engines. Most of the improvements relate to reciprocating engines, but one improvement relates to a rotary engine or turbine. This engine, or turbine, is described and illustrated in one of its "most commodious" forms by Watt in his specification. A vessel, ABDEC,

is rotatable on a pivot resting on the support J (Fig. 14), and is also supported by a collar, K, at its upper end. The vessel has a vertical partition, which divides it into two chambers, and each chamber has an aperture, R, at its upper end, which can communicate with a pipe, L (Figs. 14 and 15), conveying steam from a boiler. The rotating vessel is enclosed in a containing tank or vessel, MN, which is nearly filled with mercury, water, oil, or other liquid; and valves, F, G, are provided to allow

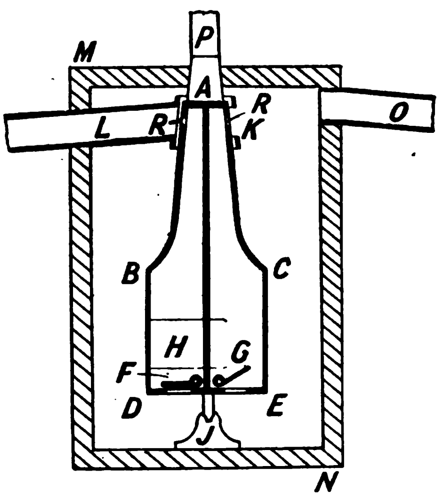


Fig 15.

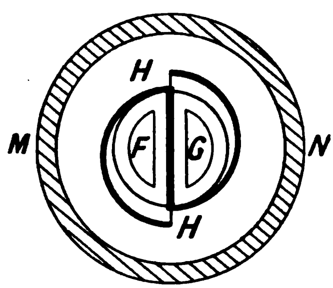


Fig. 14.

Fig. 16.

Watt's Turbine.

this liquid to enter the two chambers of the rotating vessel Fig. 16 is a sectional plan of the rotating vessel and the enclosing tank. Openings, H (Figs. 14 and 16), are provided in the sides of the rotating vessel near the bottom.

Steam enters one of the chambers of the rotating vessel through its aperture R, and forces the liquid out of the chamber into the tank by way of the hole H, the valve F or G, as the case may be, being kept closed by the pressure of the steam. The reactive force of the jet issuing from H

rotates the vessel. While the steam is entering one chamber of the rotating vessel, the steam from the other chamber is exhausting by its aperture R into the atmosphere, or into the tank to be conveyed by the pipe O to a condenser. The escape of the steam from either chamber allows the liquid in the tank to enter that chamber by the foot-valve F or G. Power for driving machinery is got from axle P. In Watt's specification drawing the rotating vessel is shown as being about 12 inches in diameter by about 30 inches high, measured to the top of the steam-pipe.

It will be seen that this turbine is the same in principle as the last-mentioned form of De Kempelen's turbine, but as Watt's specification was signed and sealed by him only about a month after De Kempelen's, and as he had been granted his patent a few months previously, it seems probable that he devised his turbine quite independently of De Kempelen.

Since the days of James Watt, a great number of patents have been granted for inventions relating to steam turbines. A selection has been made of those which the author considers most interesting and most important, but only a very small proportion of those of recent years can of course be noticed.

In 1791 **James Sadler,** an engineer of the city of Oxford, was granted a patent for an invention entitled, "An engine for lessening the Consumption of steam and fuel, in steam or fire engines, and gaining a considerable Effect in Time and Force." The drawings enrolled with the specification are here reproduced, and the inventor's "Explanation" is also given in full. The latter is as follows: "Fig. 1st (Fig. 17). The Steam generated in the Boiler A is convey'd by y^e Steam pipe B into y^e spindle of y^e rotative Cylinder C which is left

hollow for that purpose & connected with y^{e} pipe B by means of a stuffing Box at N which admits of the rotative motion of y^{e} spindle without loss of Steam, it there passes along y^{e} Arms of y^{e} rotative Cylinder nearly to y^{e} ends thereof where it meets with a jet of cold Water whereby it

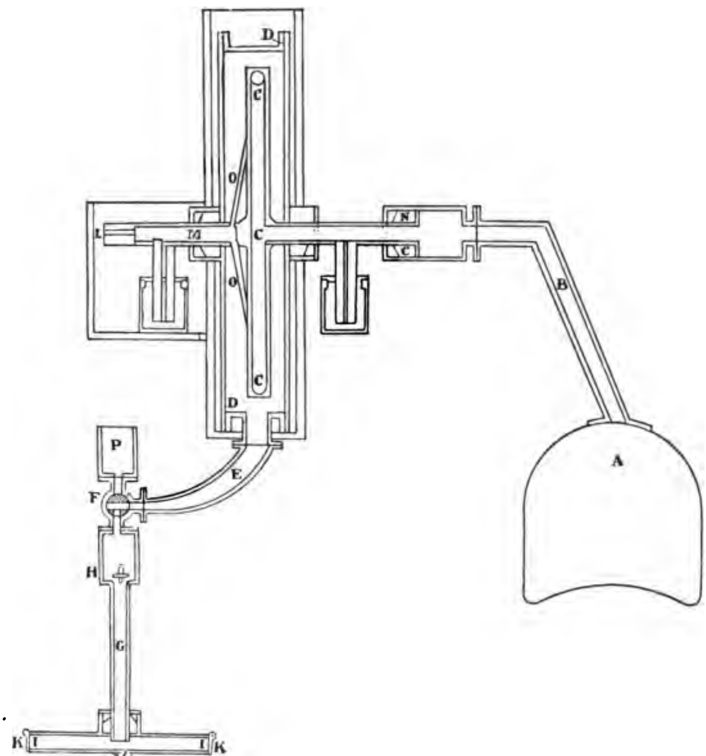


Fig. 17.—Sadler's Engine.

is condensed this jet is introduced by y^{e} small pipes OO which communicates with y^{e} spindle M which is hollow and receives y^{e} Water by a hole at L, the Water falls thro' y^{e} bottom of y^{e} case DD into y^{e} pipe E and is together with y^{e} air admitted into y^{e} pipe G thro' y^{e} Cock F and descending when y^{e} valve H is open into y^{e} pipe I which has a

rotative motion round y^e end of y^e pipe G, it is thereby ejected thro' y^e valves KK the air which is left in y^e upper end of y^e pipe G is by turning y^e cock F suffer'd to escape whilst an equal portion of Water takes its place out of the Reservoir P, Otherways y^e steam is admitted into y^e Case DD, and rushing into the Arms of y^e rotative Cylinder is therein Condensed whilst y^e external steam by its action on y^e Arm causes a rotative motion—these Arms may also be included in y^e Boiler A which will prevent the necessity of a Case. Fig. 2nd (Fig. 18) Is a Section of y^e Machine across y^e spindle of y^e rotative Cylinder before described & AA are two small pipes which convey the Cold water for injection into y^e ends of y^e Cylinder Arms at BB.

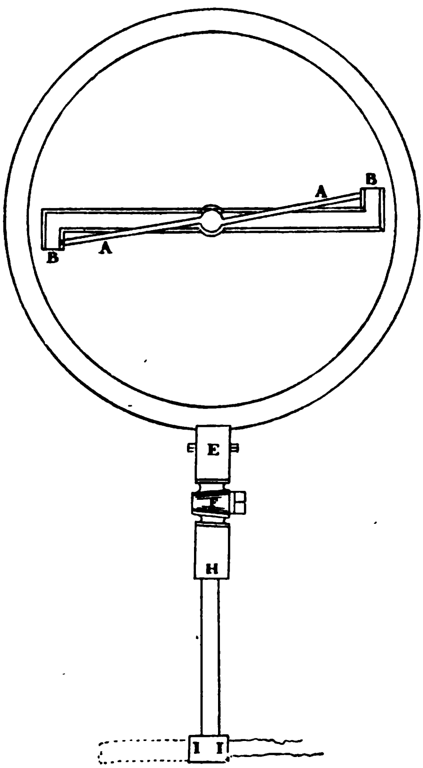


FIG. 18.—Cross-section of Sadler's Engine.

which as described before passes down y^e pipe E thro' y^e Cock F and valve H into y^e rotative arms II it is ejected from them by y^e valves KK as before described."

Noble's Patent, No. 3289 of 1809. A drawing from the specification relating to this patent is here reproduced (Fig. 19). The accompanying description is not very good, but it

is gathered that steam proceeding from the boiler A by the pipe B impinges on the "catches and ratchets" of the wheel C, and forces the wheel to rotate in the direction of the arrow. The ratchet wheel E and pawl F prevent the possibility of a contrary rotation.

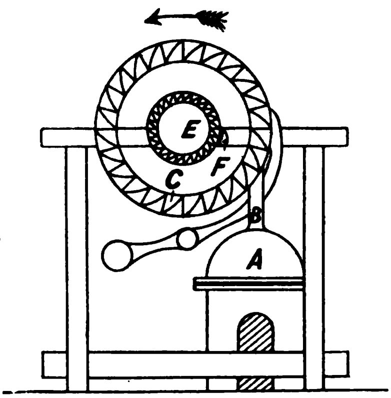


FIG 19.—Noble's Steam Wheel.

Trevithick's Patent, No. 3922 of 1815. One part of this invention consists in "causing steam of a high temperature to spout out against the atmosphere, and by its recoiling force to produce motion in a direction contrary to the issuing steam similar to the motion produced in a rocket or to the recoil of a gun." The patentee, who seems fond of firearms as similes, states that the mode of carrying this part of his invention into effect will be readily understood "by supposing a gun-barrel to be bent at about a quarter of its length from the muzzle, so that the axes of the two limbs shall be at right angles to each other, and the axis of the touch-hole at right angles to the axis of the short limb, or the limb containing the muzzle. . . . Then in the top of a boiler suitable to the raising [of] steam of a high temperature, make a hole and insert the muzzle of the gun-barrel into that hole, so that the gun-barrel may revolve in the hole steam-tight, and let the short bend of the gun-barrel be supported in a vertical position by a collar which will permit the breech of the gun-barrel to describe a horizontal circle, the touch-hole being at the side of the barrel. If steam of a high pressure be then

raised in the boiler, it will evidently pass through the gun-barrel and spout out from the touch-hole against the atmosphere with a force greater or less according to the strength of the steam, and as the steam is also exerting a contrary force against that part of the breech which is opposite to the touch-hole, the barrel will recoil, and because the other end is confined to a centre the breech end will go round in a circle with a speed proportionate to the pressure given, and may be readily made to communicate motion to machinery in general." The patentee gives this explanation "merely to convey to the mind a clear idea" of his invention. In practice, he says, he uses more than one revolving arm, and he makes the aperture through which the steam is projected capable of being increased or decreased by means of a sliding piece worked by a screw. Several other variations may also, he states, be adopted.

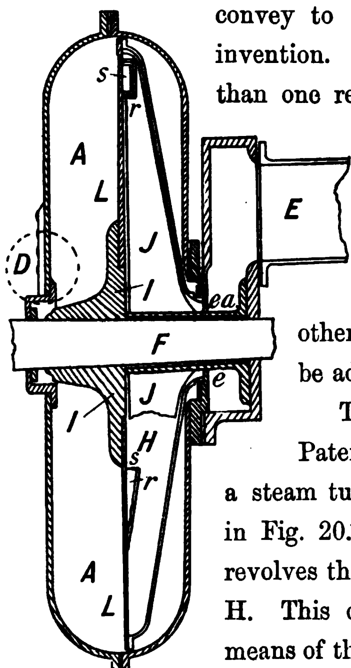


FIG. 20.—Ericsson's Turbine.

The specification of **Ericsson's** Patent, No. 5961 of 1830, describes a steam turbine, a section of which is given in Fig. 20. A is a fixed casing in which revolves the shaft F carrying the "fly-drum" H. This drum is attached to the shaft by means of the boss I and the plate L. Channels *r* are provided in the plate L, which channels open at *s* into the fly-drum. Vanes J are situated inside the fly-drum, but are not connected to it, being attached only to the fixed collar *a*. One of the channels

r is shown separately in Fig. 21. The channels are also shown in Fig. 22, which is a view at right angles to Fig. 20, and exhibits also the fixed vanes J. In Fig. 22, however, besides the channels *r* in the face of the fly-drum, channels *r′* are also shown in the periphery of the same. The steam enters the casing by the pipe D, and its action in passing into the fly-drum through the channels *r* causes the drum to rotate, while the fixed vanes J prevent the rotation of the steam which leaves the casing at *e* and passes away by the exit pipe E. The inventor states, towards the end of his specification, that the object of his invention would be equally well obtained if the steam were to travel in a reverse manner —that is, to enter the fly-drum at *e* and leave it by the channels *r*.

FIG. 21.

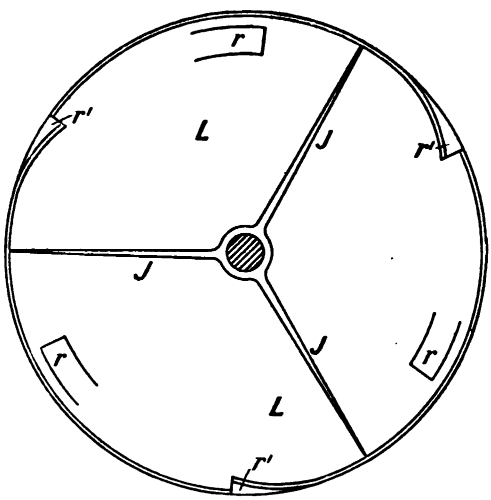


FIG. 22.—Vanes and Channels of Ericsson's Turbine.

Perkins' Patent, No. 7242 of 1836. The patentee states that in previous rotary steam-engines of the kind in which motion has been obtained by the reaction of steam-jets issuing from a rotating apparatus, the steam has been allowed to freely escape from the orifices into the atmosphere or into a steam chamber. In the patentee's engine, however, a series of

abutments, like the teeth of a ratchet wheel, are arranged in a ring for the steam-jets to impinge on.

The specification of **Pilbrow's** Patent, No. 9658 of 1843, is very interesting. The inventor seems to have experimented and theorized on the expansion and impulsive force of steam to a considerable extent. He found out, among other things, that, with a nozzle having an orifice three-eighths of an inch in diameter (the form of the nozzle is unfortunately not stated), the impulsive force of the steam issuing into the atmosphere was nearly proportional to the gauge pressure forcing the steam out. The pressures experimented with varied from 10 to 60 lbs. above atmosphere, and the impulsive force was measured "at the best distance from the orifice of the nozzle (about three-quarters of an inch)." With a gauge pressure of 60 lbs., the experimenter found that the total impulsive force (not the impulsive force per square inch) was about 14 lbs. Pilbrow calculated from this that the best velocity for the vanes of his turbine, using steam at 60 lbs. above atmosphere, would be about 1250 feet per second. He admitted that this was a very high velocity, but hoped to be able to utilize it.

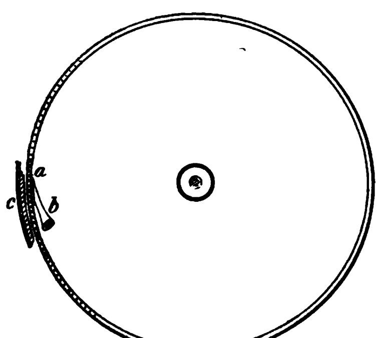


Fig. 23.—Simple Turbine of Pilbrow's

Fig. 23 shows a simple turbine wheel as proposed by Pilbrow. The steam nozzle *b* is situated inside the wheel, and projects steam against the vanes *a*, where its motion is reversed. The fixed vanes *c* lead the steam away. The change of momentum of the steam causes the wheel to rotate.

Fig. 24 shows in side elevation two such wheels mounted on the same shaft and enclosed in the same case. The vanes are set opposite ways on the two wheels, one wheel being intended for giving a reverse motion to the shaft. The pipes

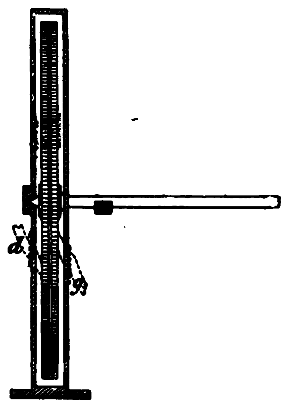


FIG. 24.—Reversing Turbine of Pilbrow's.

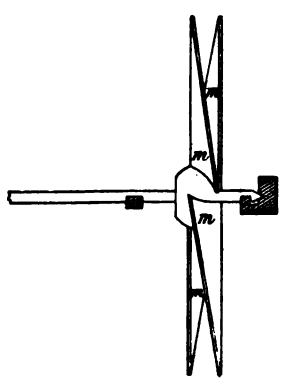


FIG. 25.—Pilbrow's Air-propeller.

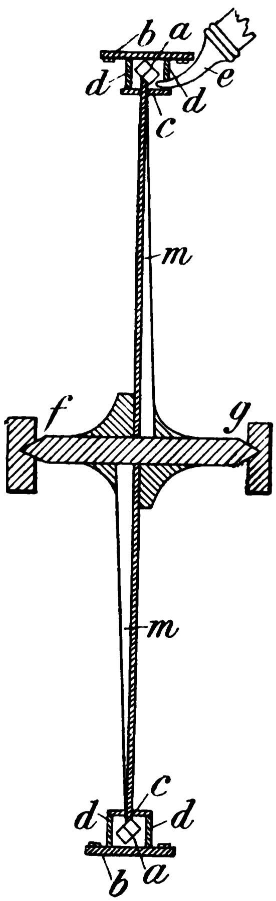


FIG 26.—Combined Steam Turbine and Air-propeller.

conducting the steam to the two nozzles are shown in dotted lines and lettered *d* and *g*. Of course only one wheel and one nozzle are used at a time.

For purposes of land locomotion the inventor proposes to use an air-propeller, as shown in Fig. 25, fixed to the shaft of the steam turbine. Fig. 26 shows in section a combined steam turbine wheel and air-propeller. *mm* are the propeller blades, such as those seen in Fig. 25, and *f, g* is the axle on which the blades are mounted. A rim, *c*, is attached to the tips of the blades, and revolves close to the edges of the annular plates *d*, which, with the hoop *b*, form

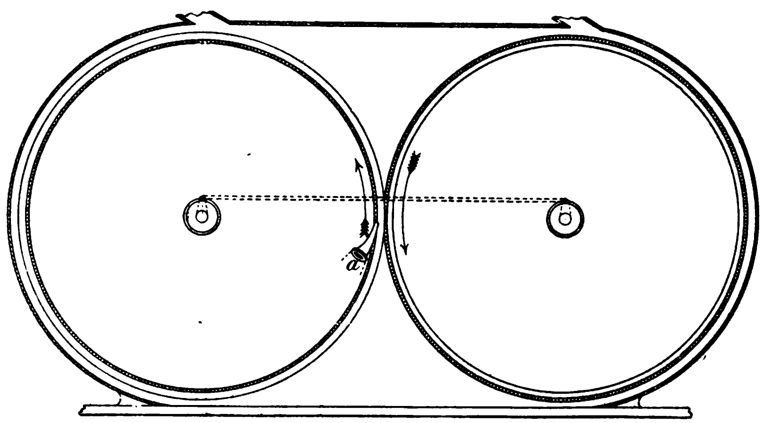


FIG. 27.—Pilbrow's Successive-expansion Turbine: Elevation.

an annular gutter. Inside this gutter, and attached to the rim *c*, are the vanes *a*, which are acted on by the steam issuing from the nozzle *e*. An eduction pipe may be provided to lead the exhaust steam away from the gutter, or this steam may be allowed to escape only at the annular openings between the fixed plates *d* and the revolving rim *c*.

In order to get a steam turbine to work efficiently at a lower speed, the inventor proposes the arrangement shown in Figs. 27 and 28. A number of wheels are placed to

rotate on two parallel axes, the rims of the wheels overlapping, as shown in elevation in Fig. 27 and in part plan in Fig. 28. The wheels are arranged as parallel-flow turbines, and the steam entering the first wheel from the nozzle *a*, passes in succession through the vanes of all the wheels.

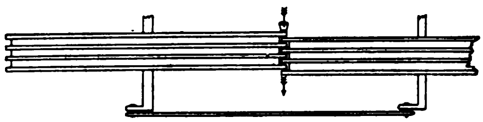


FIG. 28.—Pilbrow's Successive-expansion Turbine: Plan.

This is illustrated as regards two of the wheels by Fig. 29, which is drawn to a large scale. It will be seen that, at the parts adjacent to the nozzle, the vanes of the two sets of wheels move in opposite directions—that is, the two sets of

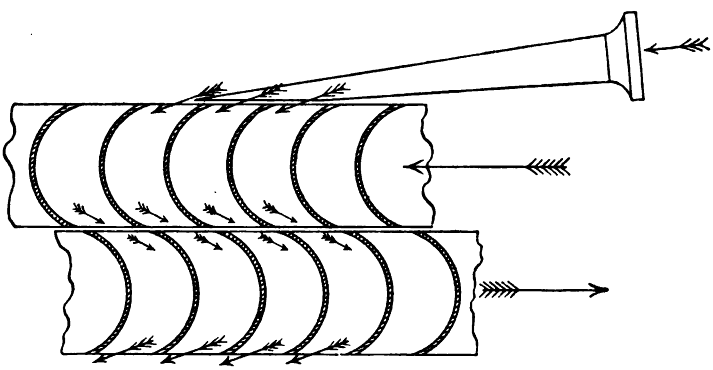


FIG. 29.—Pilbrow's Successive-expansion Turbine: Nozzle and Vanes.

wheels have similar angular velocities. The two axes may be connected by cranks and coupling-rods.

The inventor also apparently conceived the idea of reducing the vane velocity without the necessity of a second shaft by using fixed vanes or guides, for he says, "I also claim the exclusive use of curves or cavities in a stationary case to

reflect the steam back upon the wheel for a second or other number of impulses."

The inventor further describes how the power of one of his turbine wheels may be communicated to machinery by friction gearing.

The most important part of this specification is, in the

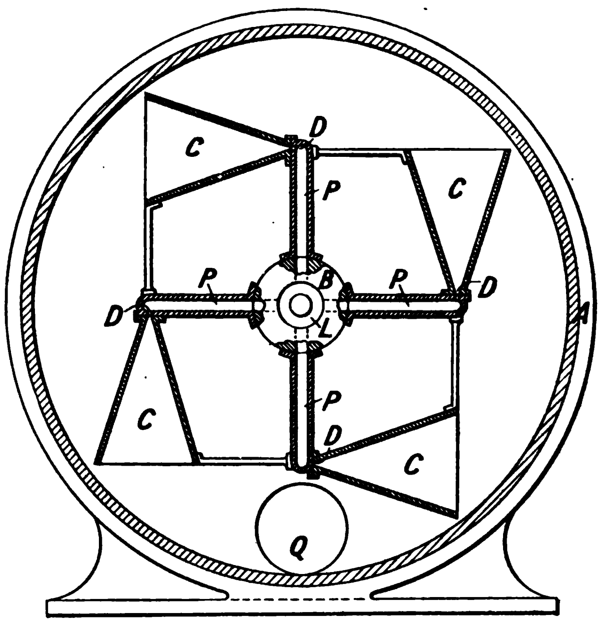


FIG. 30.—Von Rathen's Turbine.

author's opinion, the description of the method of reducing the vane velocity without losing efficiency by passing the steam through a number of rings of vanes in series. The adoption of this principle in modern turbines has contributed much to make these so serviceable.

Von Rathen's specification, No. 11,800 of 1847, contains descriptions of several varieties of rotary steam or air engines, some at least of which may be classified as turbines. Fig. 30

shows in section one variety. A is a fixed casing in which rotates the boss B, carrying the radial pipes P. At the end of each pipe P is a cone, C, whose smaller end communicates with the interior of the pipe by means of a small orifice, D. Steam is supplied to the pipes P through the hollow boss L, and escapes, after expansion in the cones, by the pipe Q, to the atmosphere or the condenser. The boss B is mounted

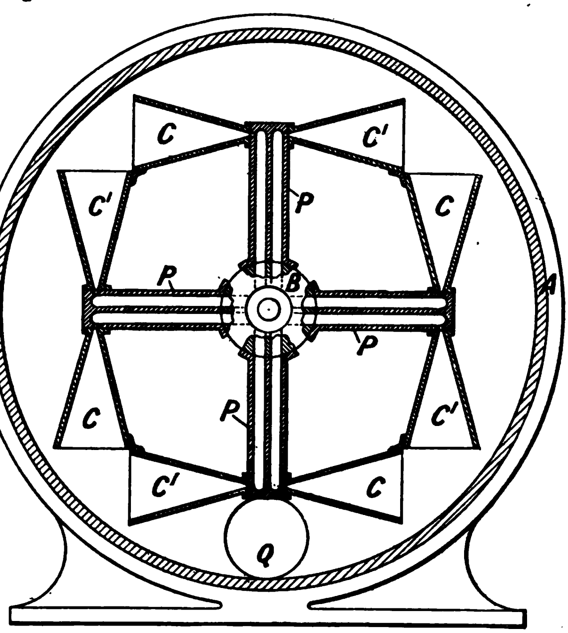


FIG. 31.—Von Rathen's Reversing Turbine.

on an axle, which passes through the flat sides or ends of the casing. To render these parts steam-tight, the inventor proposes to use metallic bushes or packings, "and rings of gutta-percha, sulphurized caoutchouc, or similar substances." Fig. 31 shows a modification of the type of engine just mentioned intended for reversing. The pipes P are here made double. One chamber of each pipe communicates with a cone,

C, while the other chamber communicates with a pipe, C′. Steam can be admitted either to the cones C or the cones C′, and the engine can, therefore, rotate in either direction. The

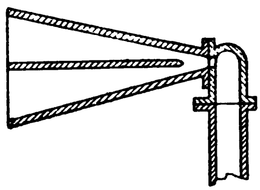


FIG. 32.

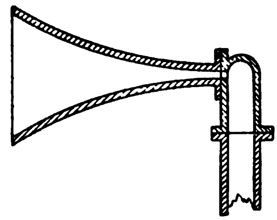


FIG. 33.

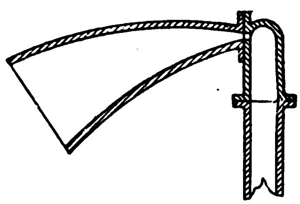


FIG. 34.

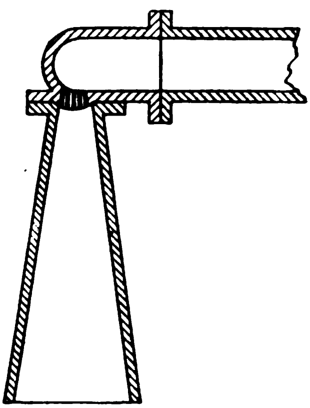


FIG. 35.

Forms of Expanding Cone or Nozzle for Von Rathen's Turbine.

inventor describes and illustrates various constructions of expanding cones or their equivalents. Some of these are illustrated in Figs. 32, 33, 34, and 35. Several other varieties of engine are described, in some of which the casing revolves as well as the boss.

In 1848 **Robert Wilson,** of Greenock, was granted a patent for improvements relating to rotatory engines. His improvements are chiefly with regard to the successive expansion of the steam. Wilson states in his specification that he is aware that, previous to his invention, steam has been employed in reciprocating engines to act successively in two cylinders, but that rotatory reacting engines have hitherto been worked only

so as to utilize the force of the steam at a single operation. The last part of the statement is not correct. Wilson seems to have been unaware of Pilbrow's compound steam turbine. But although Wilson's invention does not contain all the novelty that he attributed to it, it is nevertheless very interesting, and the specification shows that the inventor had carefully considered all the details of his engines. Some of his forms

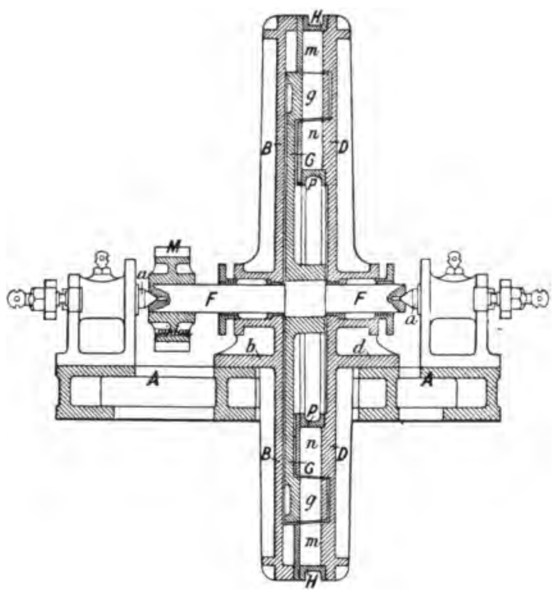


Fig. 36.—Wilson's Radial-flow Turbine with Single Ring of Moving Blades: Sectional side elevation.

and methods of construction have just recently been put in practice very much as he proposed.

One form of Wilson's turbine is shown in Fig. 36, in sectional side elevation; while Fig. 37 shows the same, half in front elevation and half in section. On a base plate, A, are mounted two discs, B and D, which are united at their circumferences by the ring H. Each of the discs has a stuffing-

box through which passes a shaft, F, adapted to rotate on conical pins, *a*. On the shaft, and between the discs B and D, is keyed a disc, G, and this disc carries a number of curved vanes, *g*, which are best seen in Fig. 37. The disc D carries a number of vanes, r^1, r^2, r^3, etc., and also (presumably) a number of blocks, M, separating chambers m^2, m^3, m^4, etc.

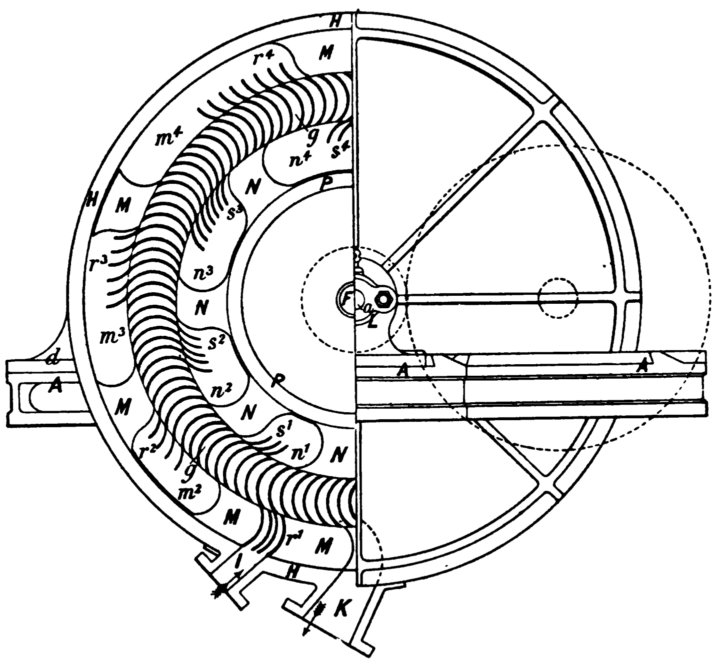


FIG. 37.—Wilson's Radial-flow Turbine with Single Ring of Moving Blades: Half section and half front elevation.

(lettered *m* in Fig. 36). The disc D also carries a number of vanes, s^1, s^2, s^3, etc., and (presumably) a number of blocks, N, separating chambers n^1, n^2, n^3, etc. (lettered *n* in Fig. 36). All the vanes are arranged in three concentric rings so that steam can pass (for example) through between the vanes r^1 and *g*, or (for example) from the chamber m^2, through between

the vanes s^1 and g to the chamber n^2, without any movement parallel to the axis of revolution of the shaft F. This is shown clearly in Fig. 36. The steam passes through the ring H at I, and between the vanes r^1 which guide it to strike the vanes g nearly tangentially to these. The steam passes through between the vanes g, enters the chamber n^1, sweeps round this chamber, and re-enters the spaces between the vanes g by way of the fixed vanes s^1. The steam then enters the chamber m^2, sweeps round it, and again enters the spaces between the rotating blades by way of the fixed blades r^2. The steam thus proceeds round the casing with a serpentine course, and eventually leaves the casing at K. The actual path of the steam will be somewhat as indicated in Fig. 38, where the solid line represents the path of the steam, and the dotted lines the internal and external peripheries of the ring of moving vanes. The stream of fluid will of course spread out in its path. The disc G, with its vanes g, presumably moves at a much lower speed than the velocity of rotation of the steam round the axis of rotation of the disc. The multiple action of the steam thus allows nearly all the energy of the steam to be conveniently used, and allows of the rotation of the

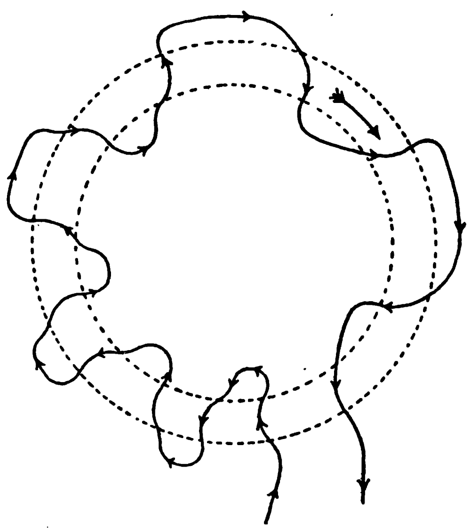


FIG. 38.

moving vanes at a speed which is small compared with the absolute velocity of the steam.

Fig. 39 shows another form of Wilson's turbine, in which the rings of blades *v*, *t*, and *s* are attached to a disc keyed on a revolving shaft, while the vanes *w*, *u*, and *g* are attached to a disc which is either stationary or is keyed

FIG. 39.—Wilson's Radial-flow Turbine with a series of Rings of Moving Blades.

to a shaft revolving in the opposite direction to the first-mentioned shaft. Steam is supplied from the boiler to the space *nn*, enters at several points the spaces between the blades, and works its way outwards through all the rings of blades. Fig. 40 shows a third form of Wilson's turbine, in which the blades *g*, *u*, and *w* are attached to and revolve with the shaft F, while the blades *v*, *t*, and *s* are fixed to the casing H, and do not move. The last two forms of Wilson's turbine are improvements on Pilbrow's device for obtaining

multiple action of the steam, and are the same in principle as successful turbines of the present day. Wilson's turbines were not intended to be mere toys. One of them is shown in the specification drawings as over 9 feet in diameter.

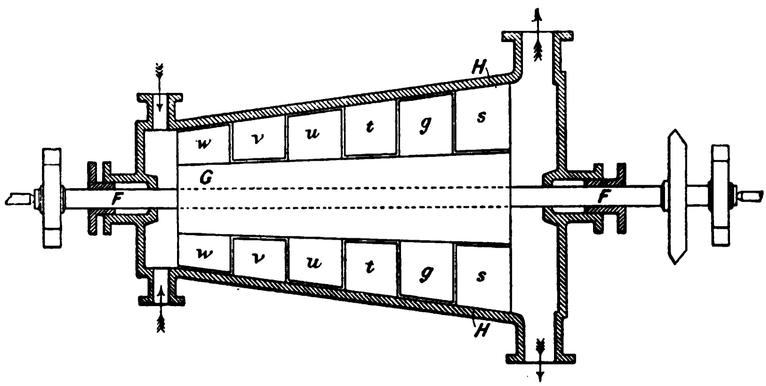


FIG. 40.—Wilson's Parallel-flow Turbine.

Fernihough's Patent, No. 13,281 of 1850. The patentee describes an apparatus in which the products of combustion from a furnace mingled with steam or water-spray are used to drive a turbine.

In 1853 the French mining engineer **Tournaire** pointed out very clearly the requisites of a successful steam turbine. Tournaire explained that elastic fluids like steam acquire enormous velocities, and that in order to properly utilize these velocities in a simple wheel, the latter would require to have an extraordinarily great speed. He further explained that the difficulty of excessive speed of rotation could be avoided by causing the steam or gas to lose its pressure in a gradual manner, or by successive fractions, and by making it act in series on a number of turbine blades. Tournaire described a machine in which there were several shafts, all of which carried pinions which geared with a common shaft

from which power could be taken. Each shaft carried a number of wheels with blades, which wheels alternated with a number of rings of blades fixed to an enclosing cylinder. The steam, after passing in series through the fixed and moving rings of blades in one cylinder, was led to the cylinder enclosing the second shaft, and so on. Tournaire recognized that very good workmanship would be required to prevent serious loss of power through leakage between the fixed and moving blades. He also recognized the difficulty with toothed wheels rotating at the necessary speeds, and suggested the use of helicoidal gearing.

The good workmanship referred to by Tournaire has contributed largely to the success of the Parsons turbine, while the helicoidal gearing is an important feature of the De Laval motor.

Patent No. 3161 of 1873, **Thomas Baldwin.** This inventor, who filed no drawings with his specification, proposed to use a machine in the form of an hydraulic turbine, in which the flow of the steam might be "inward, or outward, or parallel." He mentions that a disc may be caused to rotate by the reaction of steam-jets issuing from apertures at its periphery, or by the impulse on the disc of steam-jets issuing from apertures in the casing. The inventor proposes to employ several machines in series, the steam which exhausts from the first being employed to drive the second and then the others in succession. It is proposed that the action of the steam on the last machine should be increased by leading it therefrom to an injector or ejector where the steam would be condensed, and the kinetic energy of the condensing water would then be utilized in a hydraulic turbine or water-wheel.

Patent No. 706 of 1874, **Alexander Teulon.** This inventor

proposed to utilize the axial thrust of a steam turbine to balance the axial thrust of a screw propeller.

Figs. 41 to 46 show steam turbine details which formed the subject-matter of several letters patent granted to **John S. Raworth**, about 1894.* 1, 1^a, 1^b, Fig. 41, are ports in communication with the nozzles of a turbine, and 2 is a circular valve furnished with ports, 2^a, 2^b, 2_c, in the form of slots with circular ends. The governor is connected to the valve,

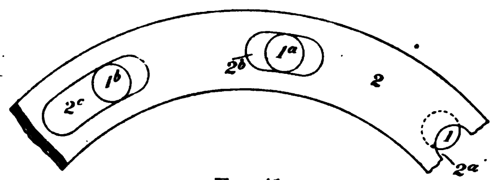


FIG. 41.

so that, when the load on the turbine falls, the valve is turned to the right, and cuts off the steam supply, first to the port 1, and then in succession to the ports, 1^a and 1^b. When the load is increased, the valve is caused to move in the opposite direction.

Fig. 42 shows a compound nozzle, which is intended to be screwed at 3 into the main steam duct. The jet of steam flowing from the main steam-duct commences to expand at 4, and, as the steam increases

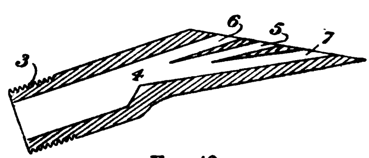


FIG. 42.

in velocity, the nozzle is developed into two or more parts, 5, 6, 7.

Figs. 43 and 44 show a device or arrangement for reducing the high speed of steam turbines by gearing to a speed suitable for ordinary industrial purposes. The turbine shaft 1 is supported in a bearing, 2, and carries a small friction

* No. 25,090, dated December 30, 1893; No. 84, dated January 2, 1894; and No. 1242, dated January 19, 1894.

wheel, 3, which gears with large friction wheels 3^a and 3^b. These large wheels are mounted on shafts, 4 and 4^a, which carry toothed pinions, 9^a and 9^b, which gear with a spur-wheel, 9, mounted on a shaft, 10, from which power can be taken. The shafts 4 and 4^a are supported in bearings in levers, 5^a and 5^b, which are pivoted at 6 and 6^a to the

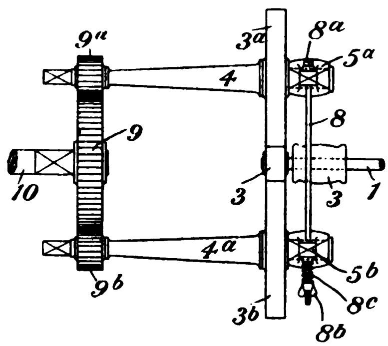


FIG. 43.

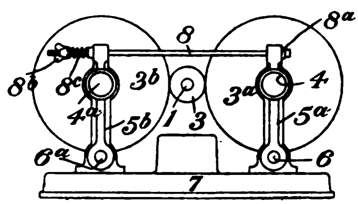


FIG. 44.

base-plate 7, and are linked together at their upper ends by the rod 8, having a head, 8^a, and a nut, 8^b. A spring, 8^c, is arranged on the rod so that, by adjusting the nut 8^b, the wheels 3^a and 3^b can be pressed against the small wheel 3 with any desired pressure.

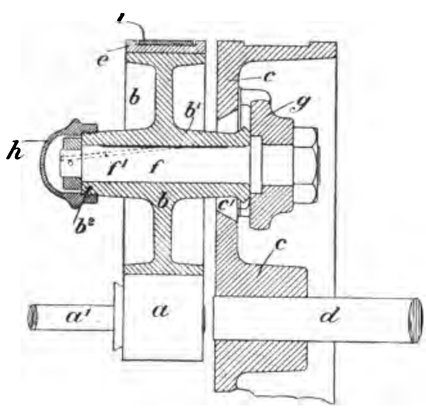


FIG. 45.

Fig. 45 shows another method of reducing the speed. The turbine shaft a' carries a pulley, a, which gears frictionally with three wheels, b, of which only one is shown. The wheels b rotate on studs, f, attached to swing-frames, g, one of which is shown separately in Fig. 46. Each wheel, b, is lubricated by means

of a channel, f', leading from an oil-chamber enclosed by the cap h screwed on the boss b^2 of the wheel. This construction prevents oil dripping on to the friction wheels. The frames g are pivoted at q^1 to the plate c, to which is keyed the power-shaft d. The frames may be weighted at q^2 to balance the weights of the studs and friction wheels. The latter are pressed against the small wheel a by a flexible band, e, which encircles the three wheels b, and is of such a diameter that it has to be sprung to extend around them. The band may be prevented from rotating by a band-brake, i.

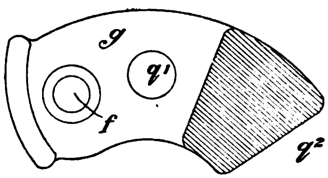


FIG. 46.

Alexander Morton, of Glasgow, made several experiments with steam turbines about 1888 to 1892. In one of his engines a series of cylinders was arranged one within the other, the ends of the whole being closed by two common discs. Steam was admitted to the interior of the inner cylinder, and expanded through nozzles into the surrounding cylinder, and this action was continued till the steam reached the last cylinder, which was in communication with a condenser. This action of the steam caused the cylinders to rotate, all moving together. No guides whatever were used during the several stages of expansion, and the engine acted wholly by reaction. Parts of three of the concentric cylinders are shown diagrammatically in Fig. 47, the nozzles also being shown. The large arrow indicates the direction of rotation of the cylinders, and the small arrows the direction of motion of the steam relatively to the cylinders.


FIG. 47.—Concentric Cylinders and Nozzles of Outward-flow Turbine of Morton's.

In another of Morton's engines (proposed, if not tried) the steam was conducted from the centre of a rotating part to the circumference by way of a number of converging channels, and was then allowed to expand in a tangential direction through

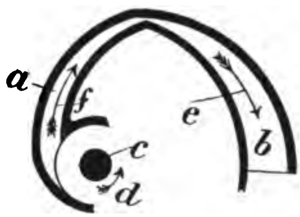


FIG. 48.—Steam Duct and Nozzle of Outward-flow Turbine of Morton's.

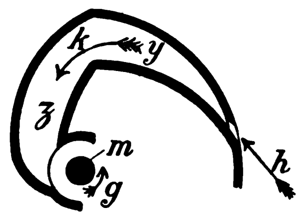


FIG. 49.—Steam Passages for Inward-flow Turbine of Morton's.

a number of diverging nozzles. Fig. 48 shows the construction diagrammatically, one converging passage, *a*, and one diverging nozzle, *b*, being shown; *c* represents the shaft which carries and is driven by the rotating parts; the arrow *d* represents the direction of rotation of this shaft; and the arrows *e*, *f* represent the direction of flow of the steam in the channel and nozzle.

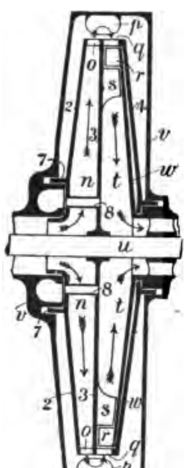


FIG. 50.—Inward and Outward-flow Turbine of Morton's.

Fig. 49 indicates diagrammatically the arrangement and form of passages, *y*, *z*, for an inward-flow turbine, the arrow *g* showing the direction of rotation, and the arrows *h* and *k* the direction of flow of the steam relatively to the rotating parts; *m* is the axis of rotation.

Fig. 50 illustrates diagrammatically part of a radial-flow turbine of Morton's, in which the steam alternately passes inwards and outwards. The arrows indicate the path of the steam, which flows freely from the centre of the rotating conical chamber *n* to the periphery of the same,

where it passes through divergent passages, *o*, of the nature of that shown at *b* in Fig. 48. It then has its motion changed by guides, *p*, and traverses divergent passages, *q*, somewhat similar to that shown at *y* in Fig. 49. The steam, continually expanding, has its motion then altered by guide-vanes, *r*, and impinges on rotating vanes, *s*. It then passes to the centre of the conical chamber *t*, where it escapes from the turbine, or is again similarly treated. The passages *o* and *q* and vanes *s* are arranged so as all to help to rotate the chambers *n* and *t* and the shaft *u*. The casing *v* is fixed, as is also the dished plate *w*, which supports the guide-vanes *r*.

The steam in the chamber *n* will press with equal intensity on the plate 2 as on the plate 3; but the steam in the chamber *t* will not press with equal intensity on the plate 3 as on the plate 4, if the fixed plate *w* be made solid. Further, there is no portion of the plate 2, and no portion of the plate 4, corresponding to the central portion of the plate 3; and, as this centre portion of the plate 3 has unequal pressures on its two sides (for the steam expands in passing from the chamber *n* to the chamber *t*), there will be a net axial pressure from left to right.

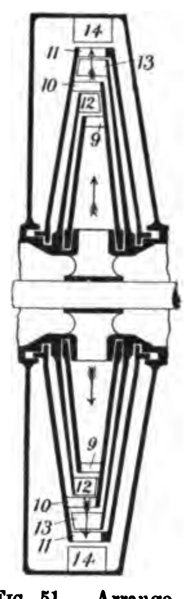


Fig. 51. — Arrangement of Vanes and Channels in Morton's Turbine.

This axial pressure is balanced by shutting off a portion of the exterior of the plate 2 from the pressure in the casing *v* by means of the ring 7, the part of the plate within the ring being subjected to the pressure in the chamber *t* by means of the tubes 8.

Another arrangement of vanes and channels is shown

diagrammatically in Fig. 51, the steam passing radially outwards as indicated by the arrows, and traversing in succession diverging passages 9, 10, 11. Guide-vanes 12, 13, and 14 receive the steam after leaving the diverging passages, and redirect its course.

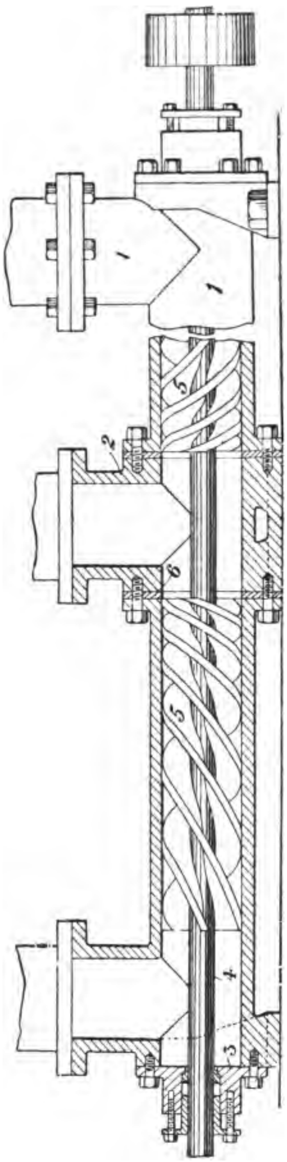


FIG. 52.—Screw Type of Steam Turbine.

Fig. 52 shows in partial sectional elevation a steam turbine of the screw type, experimented on by **Professor Hewitt.** A shaft 4 is provided in a cylindrical casing, in the ends of which are stuffing-boxes. The shaft is provided with screw-threads, 5, whose pitch increases from the centre to the ends. Steam or other fluid enters the casing by way of the branch 2, and, passing through holes in the plates 6, gains access to the helical grooves between the screw-threads. The steam leaves the casing by the branches 1 at the two ends One of the plates 6 is shown separately in Fig. 53. Professor Hewitt states that this turbine did not give good results, and that he considers that this was due to the absence of guide-plates for the steam. This is probably the case. The steam

would, no doubt, act effectively when it first struck the screw-threads; but, after it had once been deflected into a helical course, it would rush to the exhaust port, without producing much additional effect as regards rotating the shaft.

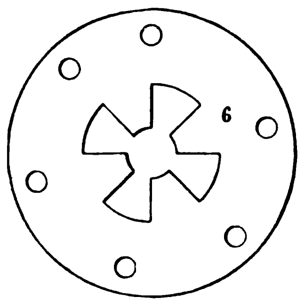


FIG. 53.—Admission Plate.

Only a small selection of inventions relating to the steam turbine could be reviewed in this chapter. Several not here referred to are described in the paper presented by Mr. Sosnowski to the International Congress on Applied Mechanics, held in Paris in the summer of 1900.

CHAPTER III.

HISTORY OF THE PARSONS STEAM TURBINE.

On April 23, 1884, the Honourable Charles Algernon Parsons filed two applications for letters patent. These were the first patents of the great inventor relative to steam turbines, although he had previously experimented with rotary engines of another type. One of these patents is entitled "Improvements in Rotary Motors actuated by elastic fluid pressure, etc." An engineer reading this specification is at once struck with the apparent practicability of the motor therein described compared with most of its predecessors of a similar type. The motor as described and illustrated shows that an immense amount of thought and attention had been spent on details —on devices for reducing cost of construction, for preventing vibration, for drawing off leaking steam, for providing efficient lubrication, etc. This attention to details has characterized the Parsons turbine throughout its life (short as yet), and probably to this is largely due the immense success of the present-day motor.

No attempt will be here made to describe in full the first Parsons turbine, as some of the details are now obsolete, but some of its interesting features are here illustrated and explained. Fig. 54 is a plan, partly in section, of the main part of the motor. A spindle, S, is formed with a central

collar, S^1, and reduced ends, S^3. On S are placed a number of rings, B, B, which are held in place between the collar S^1 and nuts S^2 screwed on the spindle. The rings are provided at their circumferences with blades, b, b^1, b^2, which are interspaced between blades, f, f^1, f^2, fixed in the inside of the turbine casing. Steam is admitted to the annular chamber g, and passed through the rings of blades in series till it reaches the exhaust ports h, h. Any steam that leaks through to the annular chambers, o, o, is led away to a chamber, P (Fig 55), where by the action of a live steam-jet issuing from the nozzle p, it is ejected through the pipe q. As the steam passes from the centre to both ends, there can be little axial thrust on the shaft, but what little does occur is balanced by the exhaust steam at the ends of the casing, the arrangement being such that a slight

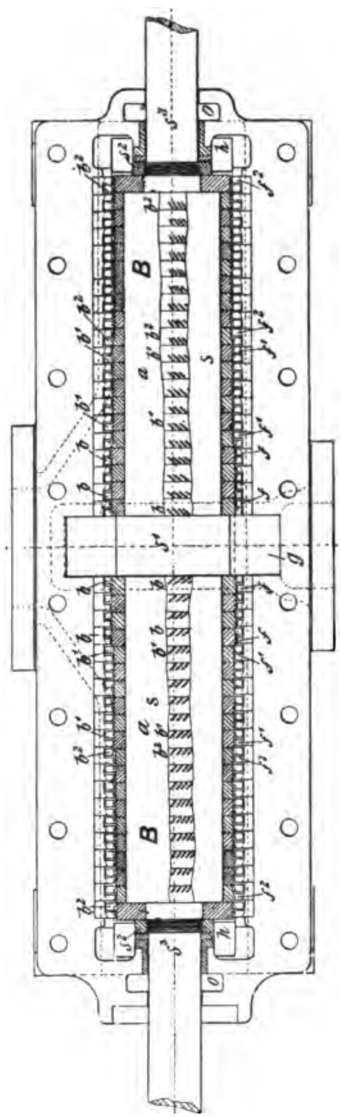


FIG. 54.—Early Parsons Turbine.

movement of the shaft to either end of the casing checks the

exhaust at that end, and so increases the back pressure. In order that the shaft and rotating parts may rotate about their centre of gravity instead of about their geometric centre when the two are not coincident, arrangements are provided for allowing the shaft a little lateral play. One of these arrangements is shown in Fig. 56, where I is a light bush enclosing

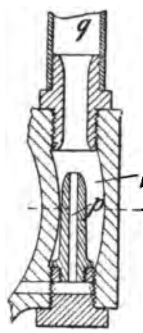


FIG. 55.—Escaped-Steam Ejector.

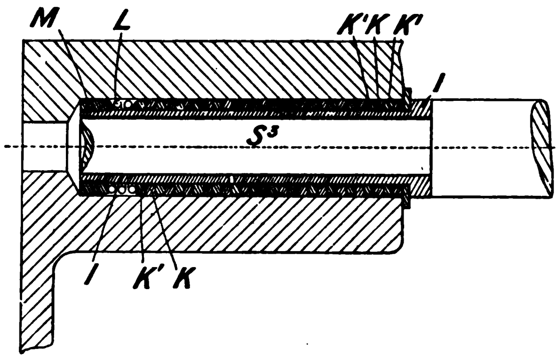


FIG. 56.—Bearing for Spindle in Early Parsons Turbine.

the shaft. Surrounding this bush are rings, K, which touch the casing but not the bush, alternating with rings, K', which touch the bush but not the casing. The nut M compresses the spiral ring L against the end ring K'. The shaft can thus move laterally a certain extent, say, one-hundredth of an inch, but this movement is resisted by the friction of the collars on one another. A system of forced lubrication is provided, and also a fan governor.

A steam turbine dynamo was constructed in 1885 by Messrs. Clarke, Chapman, Parsons and Co. Revolving at the rate of 18,000 revolutions per minute, it gave great satisfaction, and was used for several years generating current for incandescent electric lamp manufacture.

A year or two later Parsons introduced an improved

steam turbine, of which an elevation, partly in section, is given in Fig. 57. The steam entered at *a*, and passed through the rings of blades shown diagrammatically at *c* and *c'*. The fluid then passed through the rings of blades of larger diameter indicated by the letters *e* and *e'*, and then through those of still greater diameter situated at *g* and *g'*. The exhaust ends of the parts *c* and *c'* were connected by the passage *d*, which maintained an equal pressure at the two points, and the exhaust ends of the parts *e* and *e'* were similarly united by the passage *f*. The exhaust from this compound turbine was taken away from both ends by the passage *h*. Water or steam packing was provided at the places where the spindle passed through the ends of the casing, so that water or steam might be drawn into the condenser, but no air could. An annular chamber, *i* (Figs. 58 and 59), was provided round the spindle *b*

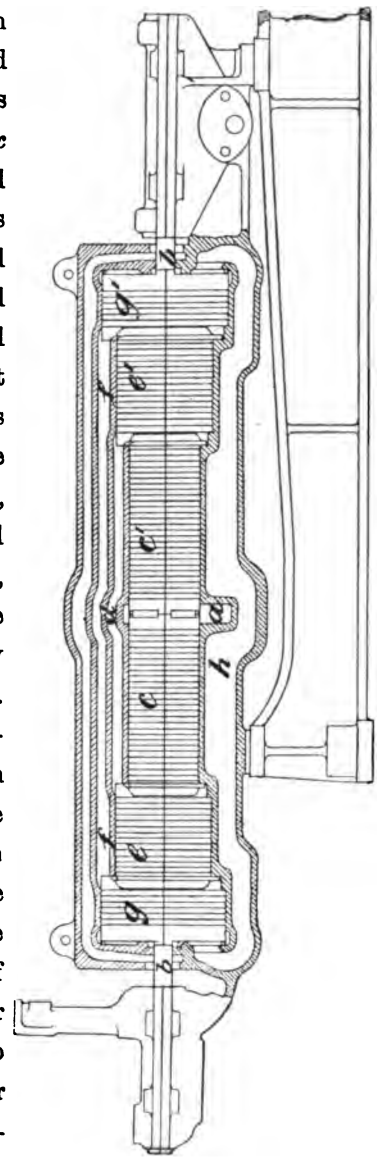


FIG. 57.—Double-ended Parsons Turbine of Increasing Diameter.

and kept supplied by the pipe *k* with water from the hot well or with steam, either at boiler pressure or partly expanded. Packing rings, *l*, *l*, *m*, were used, as shown in Fig. 58, or, when water was employed, the spindle was sometimes cut with right- and left-hand threads, as shown in Fig. 59, so that its rotation tended to repel the water leaking past.

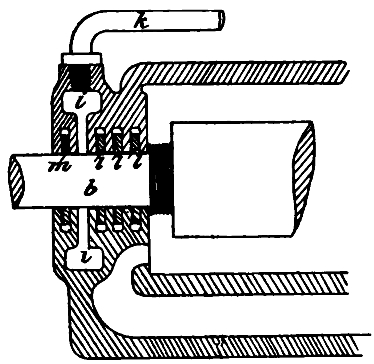


Fig. 58.

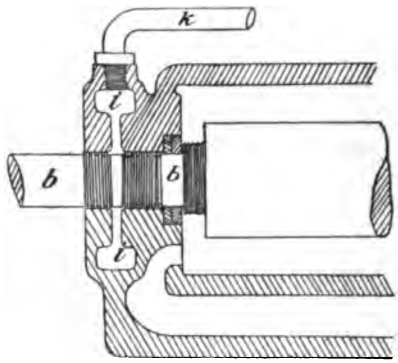


Fig. 59.

Steam or Water-packing for Spindle of Parsons Turbine.

In 1891 the first Parsons condensing steam turbine was constructed for the Cambridge Electric Supply Company by the firm of C. A. Parsons and Co., just then formed (Messrs. Clarke, Chapman, Parsons and Co. having dissolved partnership in 1889). This engine was tested by Professor Ewing and its efficiency proved to be equal to that of the best reciprocating engines of the same power.

This condensing steam turbine was followed by many others, plants being supplied to the Newcastle and District Electric Lighting Company, the Cambridge Electric Supply Company, and the Scarborough Electric Supply Company. At first the turbines had all been comparatively small, but larger machines were now made, and the increase in size,

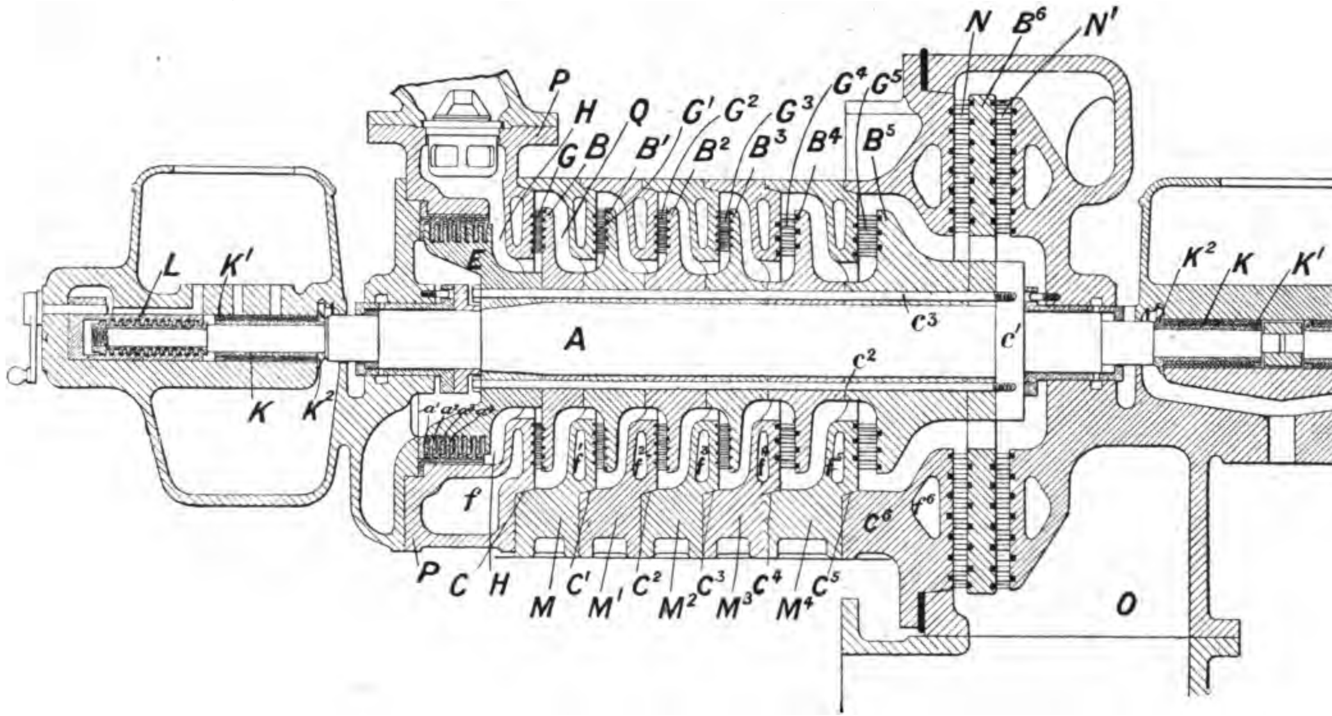


Fig. 60.—Section of Parsons Radial-flow Turbine.

together with improvements in design, led to still higher efficiencies.

Fig. 60 shows in longitudinal vertical section a radial-flow Parsons turbine. Steam is led into the annular chamber H, and passed therefrom through the fixed and moving rings of blades G, of which the fixed blades are attached to the casting P, and the moving ones to the disc B. The steam, in a somewhat expanded state, then doubles back along the passage Q, and works its way outwards again through the rings of blades G^1. The fixed blades in this case are attached to the annulus M. The action is repeated through the rings of blades G^2, G^3, G^4, and G^5. The form of these rings of blades is shown in Figs. 3–10, pp. 3–6. The final expansion of the steam takes place in the rings of blades N and N^1, and the steam then reaches the passage O and proceeds to the condenser. The method of fitting the casting P to the parts M, M^1, M^2, etc., by means of spigot and faucet joints, is clearly shown.

E is a balance piston used to balance the end pressure of the steam on the discs B, B^1, B^2, etc. This piston is provided with deep projecting flanges, a, a^1, a^2, a^3 (Figs. 60 and 61), which flanges are adapted to rotate in corresponding recesses provided in a ring secured to the casting P. The flanges are serrated on one side, as shown at b, b^1, b^2, and b^3. The resistance to the flow of steam through the tortuous passages between the fixed and moving flanges is very great, and leakage is thus reduced to a minimum. The piston E is mounted on a conical part of the spindle.

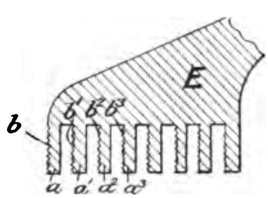


FIG. 61.

The turbine spindle A is constructed with a collar, c^1, into which are screwed long studs or pins, c^2, c^3, which pass

through holes in the turbine discs B, B^1, B^2, etc., and through holes in the balance piston E. The discs and balance piston are thus firmly held on the spindle. Live steam is admitted to the annular spaces f, f^1, f^2, etc., to reduce the condensation of the steam passing through the rings of blades.

In order to damp vibration and to allow the spindle a little transverse movement so that it may rotate about the line containing the centre of gravity of the revolving parts, the spindle is enclosed near both ends in a sleeve, K (Figs. 60, 62, 63), provided with a flange, K^2, and a collar, K^1.

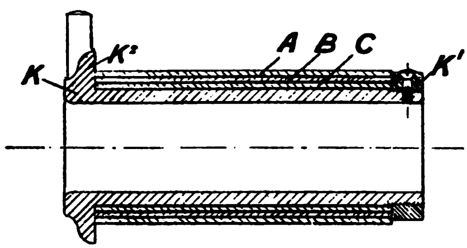


FIG. 62.

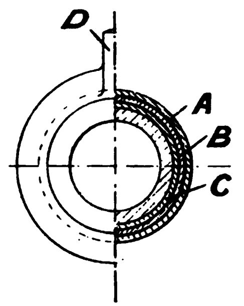


FIG. 63.

Bearing for Spindle of Parsons Turbine.

Surrounding the sleeve, and between the flange and collar, are placed three concentric tubes, A, B, and C. The tubes are bored so as to be an easy fit on each other and on the sleeve; and oil is supplied to the thin annular spaces so formed so that any transverse movement of the shaft is resisted by the fluid friction of the thin films of oil which have to be squeezed from the parts where the tubes are compressed against each other. Figs. 64 and 65 show an alternative construction, where two tubes, A and E, contain between them several segments, F, G, H, which are cut from a tube of smaller diameter so that the ends of the segments touch the

inner tube E, and the middle portions of the segments touch the outer tube A. Oil is supplied in this case also to the

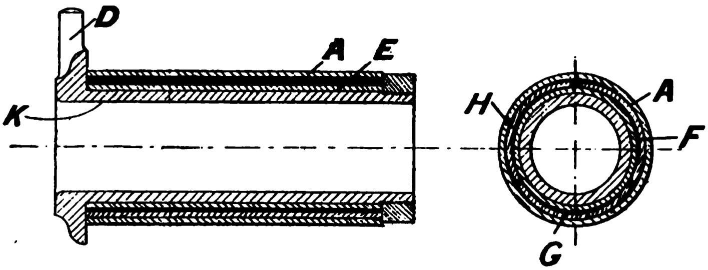


Fig. 64. Fig. 65.
Elastic Bearing for Parsons Turbine.

spaces between the tubes and sleeve, but the fluid friction is aided by the elasticity of the segments F, G, H. In both cases suitable means, such as projections D, are provided to prevent rotation of the sleeve K. The tubes are often perforated.

Any end-thrust of the spindle, due to want of perfect

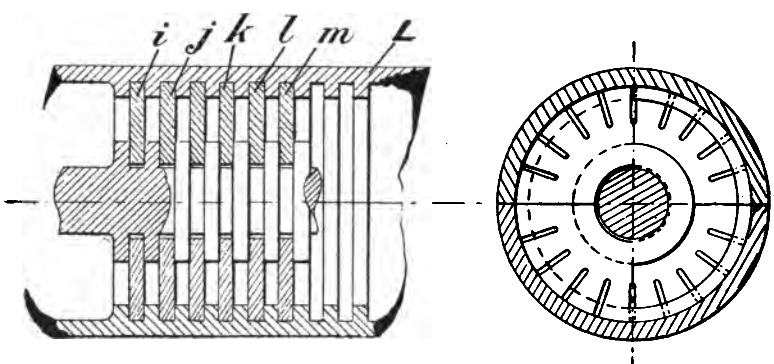


Fig. 66.—Thrust-block of Parsons Turbine. Fig. 67.—Slotted Ring for Thrust-block.

balance of the steam pressure, is taken up by the thrust-block L (Fig. 60), which is made in halves and provided with flanges and recesses to engage with recesses and flanges on the spindle. Sometimes the construction shown in Fig. 66 is

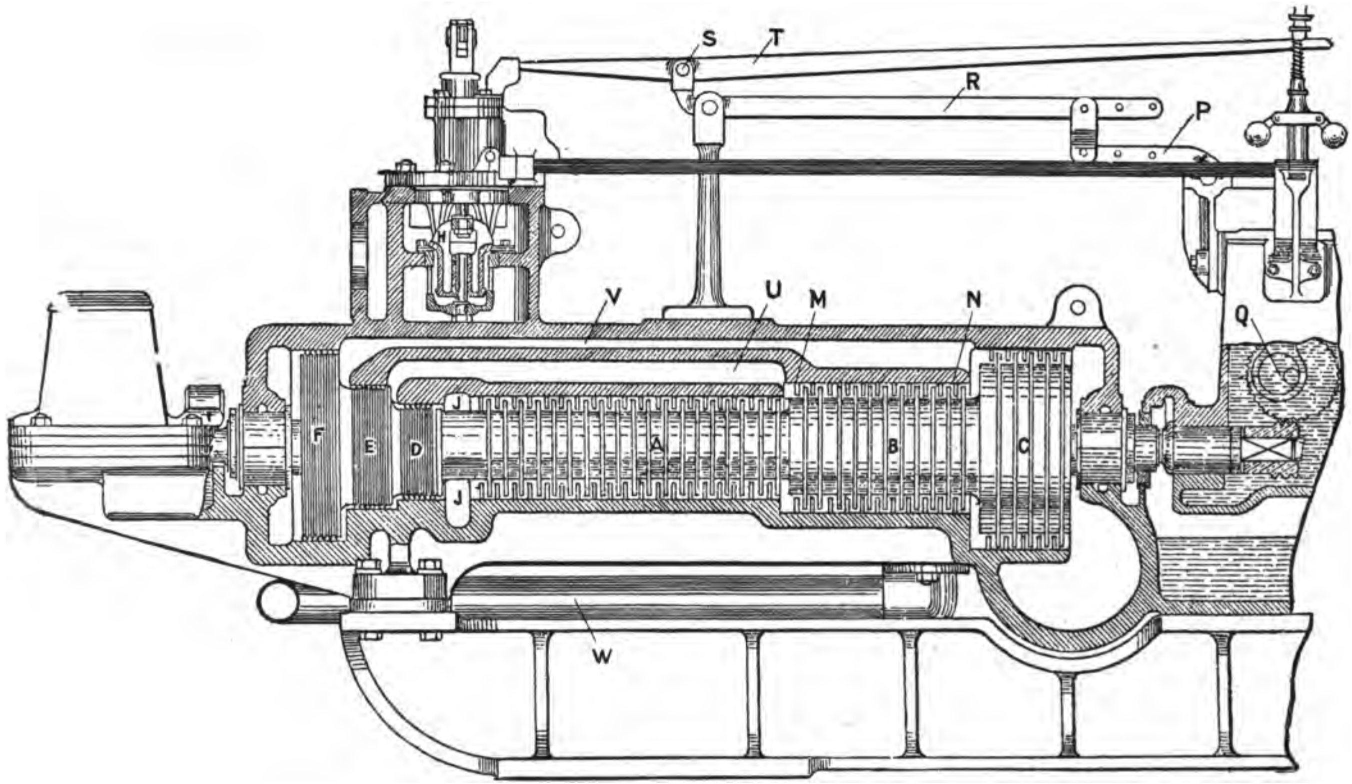


Fig. 68.—Vertical Section of Parsons Parallel-flow Steam Turbine.

adopted where rings, *i*, *j*, *k*, *l*, *m*, are used, which are separate from both block and spindle, and are of sufficient diameter and thickness to possess the requisite elasticity. The elasticity may be increased by providing slots in the rings as shown in Fig. 67; or spring washers may be inserted between the rings and the recesses for them in the block L.

All these devices for taking up end-thrust and damping vibration have been patented by Parsons.

Radial-flow turbines were constructed by Messrs. C. A. Parsons and Co. chiefly between 1889 and 1891. Parsons turbines are now usually constructed of the parallel-flow type.

Fig. 68 shows in vertical longitudinal section a modern Parsons parallel-flow turbine. Steam passes through the equilibrium valve H and enters the annular space J, from which it proceeds through the fixed and moving blades in the high-pressure cylinder, or part A; then through those in the intermediate cylinder, or part B; and then through those in the low-pressure cylinder, or part C. The arrangement and construction of the rings of blades will be best understood by referring back to Figs. 3–8. In passing through these rings of blades the steam is expanded in small steps from the initial pressure right down to the pressure in the exhaust pipe, which in a condensing turbine ought to be practically the pressure in the condenser.

In order to balance the axial thrust of the steam on the moving blades, balance pistons, D, E, F, are provided. The spaces between these balance pistons are connected by ducts, U, V, with different parts of the turbine casing, while the space beyond the largest balance piston is connected to the exhaust end of the turbine by the pipe W. Another duct of

the same nature as U and V sometimes takes the place of the pipe W.

Parallel-flow turbines are not now made double,* as shown in Figs. 54 and 57, as this necessitates the employment of a much longer shaft. Besides, as this construction really means two smaller turbines instead of one larger one, the cost is greater and the efficiency less.

Instead of making the turbine cylinder of increasing

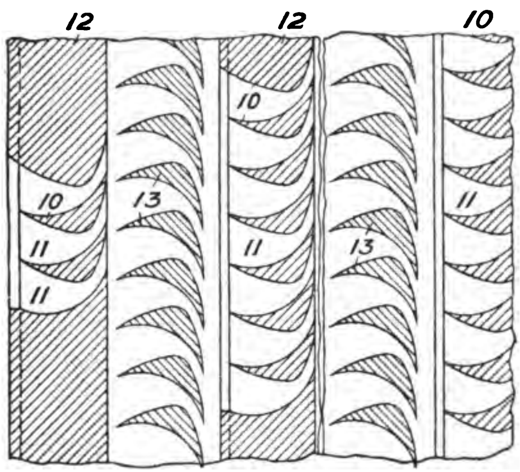


Fig. 69.—Fixed and Moving Blades of Parsons Turbine.

diameter, the fixed rings of blades at the high-pressure end may contain only a few blades, the spaces where blades are not placed being occupied by solid or hollow segments. The number of blades on the fixed rings will then increase progressively from the high-pressure end to the low-pressure end of the turbine. All the moving rings, however, are provided with blades round their whole circumferences. A section

* The author understands that the British Westinghouse Electric and Manufacturing Co., Ltd., are at present (March, 1904) constructing some large double turbines.

of the fixed and moving blades is shown drawn to a large scale in Fig. 69, where 10 represents the fixed blades, 11 the spaces between them for the passage of steam, 12 the segments occupying the remaining space of the fixed rings, and 13 the rotating blades.

When the turbine is increased in diameter by steps as shown in Fig. 68, it is desirable that the area of section for the passage of the steam shall increase continuously. Messrs. C. A. Parsons and Co. therefore construct the fixed blades at the high-pressure end of each cylinder or part of the turbine with narrower exit openings than at the low-pressure end of the same cylinder or part. Thus the blades at the high-pressure end, M, of the cylindrical part B of the turbine shown in Fig. 68 are arranged with wider escape openings than at the low-pressure end, N, of the same part B.

The angular velocity of a Parsons turbine depends on the initial and terminal pressures, and on the number and diameter of the rings of blades. An idea of the usual speeds of rotation is given by Table I., which has been compiled from turbines now running:—

TABLE I.

SPEEDS OF ROTATION OF PARSONS TURBINES (CONDENSING).

Power of turbine in kilowatts.	Steam pressure in lbs. per square inch.	Revolutions per minute.
32	125	5000
75	125	4000
150	150	3500
250	150	3000
500	180	2500
1000	200	1800
1500	200	1500
2000	200	1200

By providing more rings of blades the number of revolu-

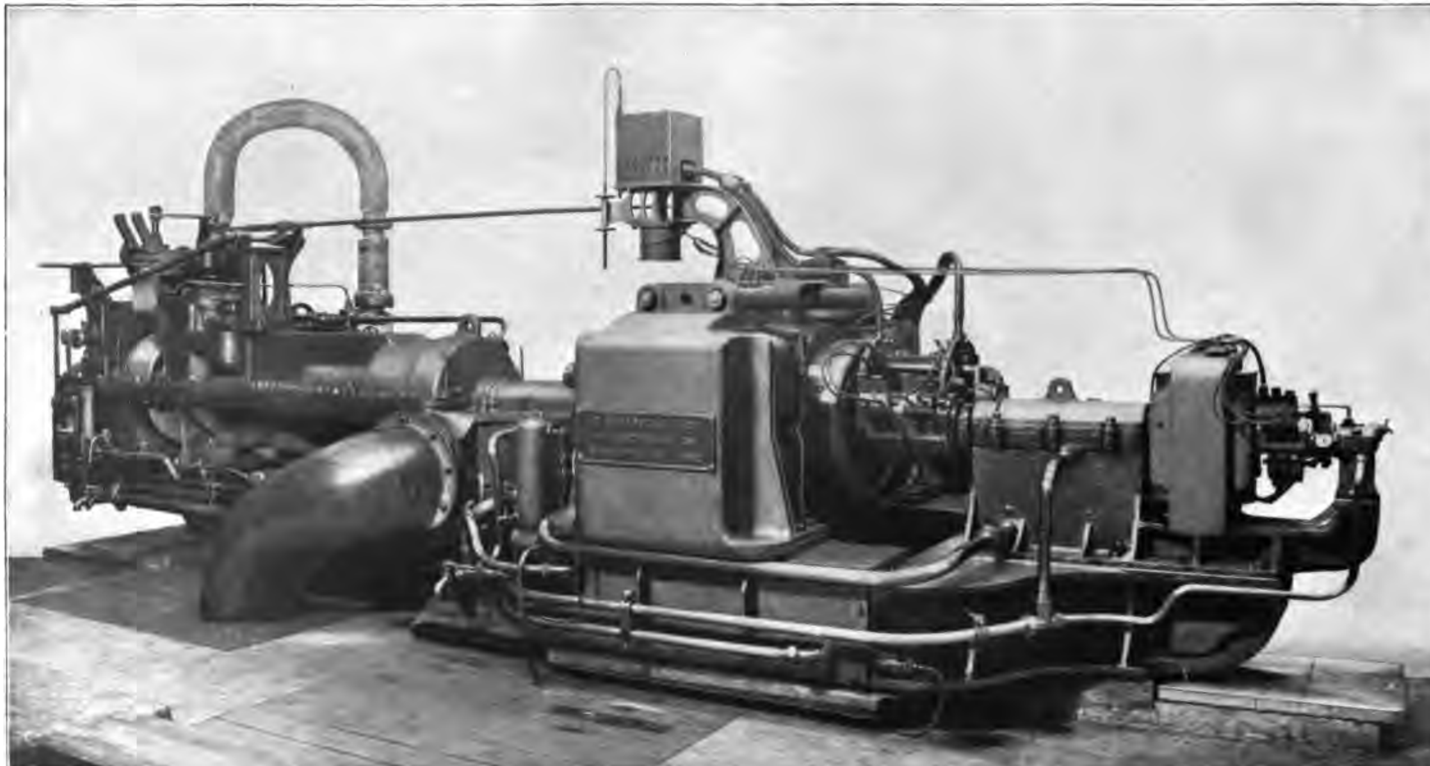


PLATE II.—PARSONS STEAM TURBINE COUPLED TO 500-KILOWATT ALTERNATOR

tions per minute can be diminished when required. A turbine can also be arranged to suit the steam pressure without altering the number of rings of blades by changing the length of these. Lengthening the blades gives more area for the passage of steam, and thus permits a lower steam-pressure to be used, while shortening the blades reduces the area, and thus allows of a higher steam-pressure.

Plate II. shows a Parsons steam turbine coupled direct to a 500-kilowatt alternator. It is installed in the Forth Banks Station of the Newcastle and District Electric Lighting Company, where it runs at 2400 revolutions a minute.

It is interesting to know that Mr. Parsons has tried a reaction steam wheel on the principle of Hero's engine. A turbine of this nature was constructed at Heaton Works, Newcastle-on-Tyne, and was provided with two arms of elliptical section (to minimize resistance), and slots at the ends of the arms. Rotating at 5000 revolutions per minute, with a steam pressure of 100 lbs. per square inch and 27 ins. of vacuum, it gave out 20 brake horse-power with a steam consumption of 40 lbs. per B.H.P. hour.

CHAPTER IV.

POINTS OF RESEMBLANCE AND DIFFERENCE BETWEEN THE STEAM TURBINE AND OTHER MOTORS.

THE action of the steam turbine depends on the conversion of the heat energy of the steam into kinetic energy, and then in the transference of this kinetic energy from the steam to the rotating parts of the turbine. The latter part of the action is thus in principle much the same as that of the water turbine, but the former part has no parallel in the hydraulic motor. In a water turbine the fluid is practically at constant volume and at constant temperature, and its kinetic energy is gained at the expense of potential energy due to pressure or position. On the other hand, when steam is used, this fluid varies in volume within very wide limits. Thus, 141 cubic feet of saturated steam at 200 lbs. pressure absolute produces 1647 cubic feet at atmospheric pressure, and this produces only 1 cubic foot of water when condensed. If the 141 cubic feet of steam at 200 lbs. pressure were expanded adiabatically till the pressure fell to 0·6 lbs. abs., then 25·5 per cent. of the steam would be condensed, and the volume of the steam and water would be 25,500 cubic feet. These volumes are represented graphically in Fig. 70, p. 56. The temperature of the steam varies also, and care has to be taken to prevent, as far as possible, loss of heat by radiation, a point that does not call for attention with a water turbine.

Another important point of difference between the steam turbine and the water turbine is the immense velocity of the fluid in the former compared with the latter. In a water turbine working under the large head of 150 feet, the velocity of the fluid entering the wheel is about 96 feet per second.

In steam turbines a fluid velocity of 2000 to 3000 feet per second is common. The reason for high speeds with steam can easily be seen. A cubic foot of water having a velocity of 96 feet per second has a kinetic energy of about 9000 foot-lbs.* A cubic foot of dry saturated steam at 50 lbs. pressure absolute has, however, so small a mass that, in order that it may have the same kinetic energy, it must have a velocity of about 2200 feet per second.* These differences in the physical properties of steam and water necessitate great differences in the construction of steam turbines and water turbines. It should also be noted that all friction in a water turbine means loss of energy; but that in a steam turbine the heat generated by the friction may serve to heat the fluid, and thus in great part restore the energy absorbed. This will be referred to again.

Comparing a steam turbine with a reciprocating engine, we find that, although the greatest possible efficiency of a heat engine comprising boiler, steam engine, condenser, etc., is the same whether the steam engine be turbine or reciprocating, and is represented by Carnot's formula $\frac{T_1 - T_2}{T_1}$, the causes which reduce this efficiency below this maximum are largely different

* Kinetic energy of 1 cubic foot of water $= \frac{mv^2}{2} = \frac{62\frac{1}{2} \times 96^2}{2 \times 32 \cdot 2} = 9000$ foot-lbs. approx.

Kinetic energy of 1 cubic foot of dry saturated steam at 50 lbs. pressure absolute $= \frac{mv^2}{2} = \frac{0 \cdot 12 \times 2200^2}{2 \times 32 \cdot 2} = 9000$ foot-lbs. approx.

in the two cases. One of the greatest losses in the reciprocating engine is due to the alternate contact of the inside of the cylinder with the hot steam and with the comparatively cold exhaust. The cylinder walls rob the entering steam of much of its heat energy. Some of this energy may be

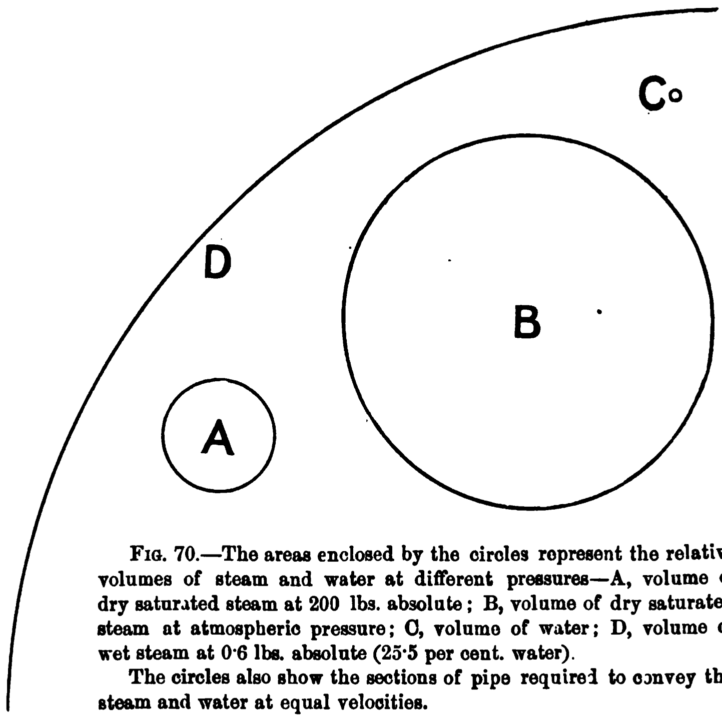


FIG. 70.—The areas enclosed by the circles represent the relative volumes of steam and water at different pressures—A, volume of dry saturated steam at 200 lbs. absolute; B, volume of dry saturated steam at atmospheric pressure; C, volume of water; D, volume of wet steam at 0·6 lbs. absolute (25·5 per cent. water).

The circles also show the sections of pipe required to convey the steam and water at equal velocities.

recovered by the steam at a later part of the stroke, but a great part is given up to the exhaust, and, unless it can afterwards be utilized, is lost. There is no such loss with the steam turbine, as the steam passes constantly in the same direction, some surfaces of the turbine making contact with the entering steam and some with the exhaust, but none with both.

Another loss which is sometimes thought to be considerable occurs with reciprocating engines using slide-valves or their equivalents, and consists in steam leaking past the valve to the exhaust. This, of course, cannot happen in a steam turbine where no slide-valve or its equivalent exists.

Another great source of loss with reciprocating engines is due to friction. This friction sometimes absorbs more than a quarter of the total I.H.P. of the engine. Except for the friction in the bearings of the shafts, the friction in a steam turbine is of a totally different nature from that in a reciprocating engine. It consists in the friction of the steam against itself and against the surfaces of the turbine, and the friction of the water carried by or deposited by the steam.

The friction between the rotating parts of the turbine and the fluid in which they revolve is very great at high speeds, even if the fluid be dry steam at a low pressure. If the steam be wet and the pressure high, the friction at extreme velocities becomes enormous.

A large amount of friction may also be produced in certain kinds of steam turbines by water deposited on the fixed parts and coming in contact with the rotating parts. Even if steam be perfectly dry when passing through the turbine stop-valve, it will naturally tend to get wet when it expands doing work; and it *will* get wet unless it is initially superheated, or other means are adopted to prevent condensation.

The use of superheated steam increases the efficiency of a steam turbine to a great extent—much more than can be accounted for by thermo-dynamic reasons—and in all cases the chief cause of the improvement is probably the reduction of friction.

It should be borne in mind, however, that in a multiple

expansion turbine, like a Parsons, heat produced by friction raises the temperature of the steam and increases its energy; and, as this energy is utilized at a later stage, the heat is not altogether lost.

With the reciprocating engine, although the friction of the piston in the cylinder and of the slide-valve or other valve in the steam-chest may heat the steam, yet, as the exhaust steam receives part of this heat, and as there is much friction caused by other parts than the piston and valve, we may safely assert that in a reciprocating engine almost the whole of the heat caused by friction is lost.

Another advantage which the steam turbine possesses over the reciprocating engine is that, with the former, there is no internal lubrication required. The fact that the steam turbine can take steam without any lubricant whatever is doubly advantageous. In the first place, the exhaust steam is absolutely free from oil, so that the water from the hot well can be directly returned to the boiler without the use of an oil filter, and without any danger of the boiler suffering from a deposit of grease in it. The second advantage arises when superheated steam is used. When this is employed in reciprocating engines, there are difficulties with regard to internal lubrication; and there is also the danger of piston and valves sticking, unless properly and carefully designed, owing to difference of expansion of different parts of the engine. With the steam turbine no lubricant is required to be added to the steam, and the danger of harm arising from unequal expansion is not as a rule great. It should be noted that, although both turbines and reciprocating engines improve in efficiency by superheating the steam, the reasons for the superheating are not altogether the same. The reciprocating engine gains

chiefly (or at least largely) by the reduction or abolition of initial condensation. This cannot be the chief reason with the steam turbine; but the gain in economy in the steam turbine by superheating will be discussed later on.

The steam turbine benefits more than the reciprocating engine from a good vacuum in the condenser. (Tables showing the effect of the state of the vacuum on the steam consumption of Parsons turbines are given in Chap. X.) Decreases of pressure below 5 lbs. absolute mean large drops of temperature in the case of saturated steam, and therefore there is a great thermo-dynamic advantage in having a low condenser pressure in a steam-engine. In the case of a reciprocating engine, however, this thermo-dynamic advantage is partly neutralized by the increased initial condensation due to the lower temperatures of the surfaces with which the steam entering the cylinder comes in contact. The increase in efficiency obtained by improving the vacuum is therefore only due to the difference of these two effects. In the case of the steam turbine, however, there is no such initial condensation, and consequently this type of engine gains largely by improvement of the condenser vacuum. Another reason for the comparatively small gain in efficiency by increase of vacuum in the reciprocating engine is the impossibility in most cases of taking full advantage of the vacuum by expanding the steam in the cylinder down to the condenser pressure without unduly increasing the bulk of the engine and diminishing its mechanical efficiency.

A source of loss with the steam turbine which does not occur with the reciprocating engine is caused at the parts where the shaft leaves the case. At high speeds of rotation difficulties obviously occur with packing such as is used in the piston-rod glands of a reciprocating turbine. In the Parsons turbine

no packing is used, but a special device is employed which will be described hereafter; with this device very little loss is said to occur.

It should be noted in comparing the driving of alternators by steam turbines and by reciprocating engines that, while the same percentage variation of speed means the same percentage variation of periodicity, a drop (or rise) of, say, 5 revolutions per minute in the one case, does not mean the same variation of periodicity as in the other case; for the number of alternations per revolution in the turbine-driven alternator is, owing to the speed of rotation, less than in the alternator driven by the reciprocating engine, the mean periodicity being the same in both cases.

As practically all the motion in a steam turbine (other than that of the steam) is rotary and constant, there is practically no vibration. This allows foundations to be very light. The lightness of the foundations required allows a steam turbine to be very quickly installed and set running; and this has proved extremely useful in many cases. It is also a very important consideration as regards cost, the saving on this score by using steam turbines in place of reciprocating engines being with large powers very great. The absence of vibration also allows a steam turbine to be installed in situations where a reciprocating engine would be impossible.

As regards cost of upkeep, reciprocating engines of different makes vary very much, but very few, if any, can rival a good steam turbine. The nature of the steam turbine lends itself to long life and small wear and tear.

CHAPTER V.

VANES AND VELOCITIES.

LET us now consider the form of the vanes or blades and the speed of rotation of a steam turbine, and, in the first instance, it may be advisable to deal with turbines generally.

As we shall be using the terms "absolute velocity" and "relative velocity" with respect to the motion of the fluid, it will be better to state here that by absolute velocity is meant a velocity which would be absolute if the turbine casing or frame were at rest. A turbine may be on board a ship, and therefore have the velocity of the ship, and even when on land and what we call fixed, it nevertheless has the velocity of the earth. It is convenient, however, to neglect these velocities of the ship and the earth and such-like, and speak of the velocity of a revolving part of the turbine or of the operating fluid as *absolute,* when we mean that such a velocity would be absolute if the casing or frame, or fixed parts of the turbine had no motion. We shall speak of velocities as *relative* only when they are relative to a "moving" part of the turbine. To illustrate what is meant, let X (Fig. 71) be part of a turbine wheel moving with an absolute velocity, W, as shown by the arrow. Let V be the absolute velocity of a jet of fluid. Then the velocity of the fluid relatively to the turbine will be obtained by making $QB = W$, and completing the

parallelogram APBQ, when PB will represent the velocity of the jet relatively to the wheel. This relative velocity PB is the velocity which the jet would have if a velocity were imparted to both the wheel and the jet of an amount sufficient to render the net velocity of the wheel equal to zero. Now, a velocity which would render the net velocity of the wheel

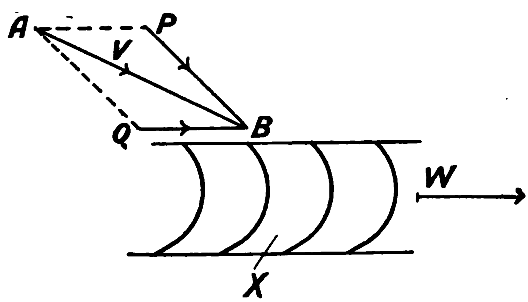


FIG. 71.

equal to zero would be equal and opposite to W. Therefore, combine this velocity with V, and the velocity PB is obtained. Or we may define the velocity of the jet relatively to the wheel as that velocity which, combined with the velocity of the wheel, produces the absolute velocity of the jet. Now, PB represents the velocity which, combined with the velocity of the wheel, produces the absolute velocity represented by AB. Therefore, PB represents the velocity of the jet relatively to the wheel.

In Fig. 72, let V = the absolute velocity of the fluid impinging on the blades or vanes of a turbine; let W = the velocity of the turbine vanes. Then R, the velocity of the fluid relatively to the turbine, can easily be determined. If the course of the fluid is not to be abruptly altered, it is necessary that the vanes where the fluid enters should be parallel to the line of R, and this is usually the case where possible. If the

sectional area of the stream or jet of fluid between two vanes is maintained constant and the volume of the fluid remains constant, then the velocity of the fluid relatively to the vanes will also be constant in magnitude. Let r represent the velocity of the fluid, leaving the ring of blades relatively to the blades. Then $r = R$ in magnitude: the direction only is altered. A'BCDEF represents the path of the fluid relatively to the blades. That, however, is not the actual or absolute path of the fluid, for the blades themselves have a velocity equal to W. If we combine the velocity W with the relative velocity of the fluid at any point, we get the absolute velocity. Thus at C the absolute velocity of the fluid is represented by GC, at D the absolute velocity of the fluid is represented by HD, and at E the absolute velocity of the fluid is represented by EK. The actual or absolute velocity of the fluid will be in the line ABJLM. EL is the distance through which the blades move while the fluid is moving between the blades from B to E.

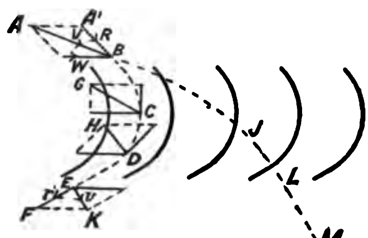


Fig. 72.

The absolute velocity of the fluid when enclosed by the vanes is not important, but the absolute velocities when entering and leaving the rings of vanes are important, as the kinetic energies of the fluid when entering and leaving the rings of vanes are proportional to the squares of these velocities. Let v be the absolute velocity of the fluid when leaving the ring of vanes. Then the kinetic energy given up by the fluid to the turbine will be proportional to $V^2 - v^2$ and the efficiency, neglecting frictional losses, will be $\frac{V^2 - v^2}{V^2}$.

It will be seen that the angle of the vanes, except at the points of entrance and exit, cannot effect the efficiency except through increasing or diminishing the frictional losses. By forming the vanes with a smooth gradual curve, and with the tangent of each at the point of entrance parallel to the relative velocity of the fluid at that point, the frictional velocities may in most cases in an hydraulic turbine be reduced to an almost inappreciable amount. The question of friction in a steam turbine is more difficult.

It is obvious that it will be desirable to have v as small as possible. Now, with a given velocity V, the smallness of v depends upon the velocity W of the vanes, and on the angles of the vanes at the points of entrance and exit of the fluid.

In Fig. 73 let *ab* represent V in magnitude and direction,

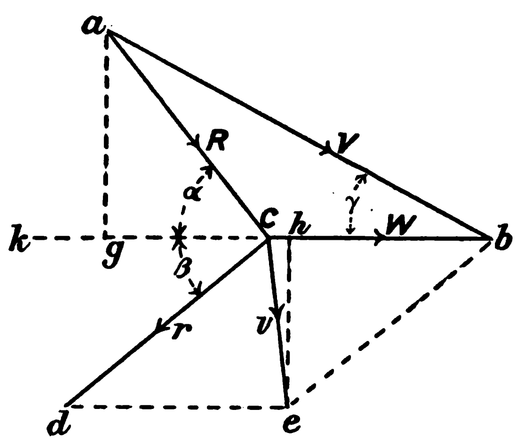


FIG. 73.

and *cb* represent W in magnitude and direction: then *ac* represents R in magnitude and direction. Let *cd* represent r in direction; then, if *cd* equals *ac*, *cd* will also (neglecting friction) represent the magnitude of r. As before stated, it is generally

advisable that the vanes at the point of entry shall be parallel to R. Therefore, if bc be produced to k, the angle ack, or α, will represent the angle of the vanes at the point of entrance, and the angle at dck, or β, will represent the angle of the vanes at the point of exit. By completing the parallelogram $cbed$, we obtain ce, which represents v in magnitude and direction.

Draw ag and eh perpendicular to bk.

$$\text{Then } ce^2 = cb^2 + eb^2 - 2bc \,.\, bh$$

$$\text{Now } eb = cd = ac$$

$$\begin{aligned}
\text{Therefore } ce^2 &= cb^2 + ac^2 - 2bc \,.\, bh \\
&= ab^2 - 2bc \,.\, cg - 2bc \,.\, bh \\
&= ab^2 - 2bc(cg + bh) \\
&= ab^2 - 2bc(ac \cos \alpha + eb \cos \beta) \\
&= ab^2 - 2bc(ac \cos \alpha + ac \cos \beta) \\
&= ab^2 - 2bc \,.\, ac(\cos \alpha + \cos \beta) \quad . \quad . \quad . \ (1)
\end{aligned}$$

$$\text{Therefore } v^2 = V^2 - 2bc \,.\, ac(\cos \alpha + \cos \beta) \quad . \quad . \quad . \ (2)$$

It is therefore evident that with a given initial absolute velocity of the fluid, v^2 will be the smallest when $2bc \,.\, ac(\cos \alpha + \cos \beta)$ is greatest. It can be seen that this will occur when α and β are each equal to zero, and when $bc = cg$, which in this case will equal ac. W would then equal $\frac{1}{2}$V, and the vanes would be as shown in Fig. 74.

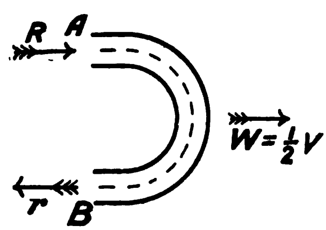


FIG. 74.

If V = velocity at A, the velocity W of the vane should be $\frac{1}{2}$V. The velocity R of the fluid relatively to the vane at A would therefore be $\frac{1}{2}$V. Therefore the velocity r of the fluid relatively to the vane at B would also be $\frac{1}{2}$V, and therefore the absolute velocity of the fluid at this point would be zero.

Not only when the angles α and β are both equal to zero, but in any case when α and β are fixed, and V is also fixed in

magnitude, it can be seen from equation (2) that v^2 is least, when $ac.bc$ (Fig. 73) is a maximum. Since the area of the triangle $abc = \frac{1}{2}ac.bc \sin a$, and also equals $\frac{1}{2}ab.cm$ (Fig. 75) where cm is perpendicular to ab, it follows that—

$$\frac{ab.cm}{\sin a} = ac.bc$$

Therefore v^2 is least when $\frac{ab.cm}{\sin a}$ is a maximum.

But ab and $\sin a$ are both constant.

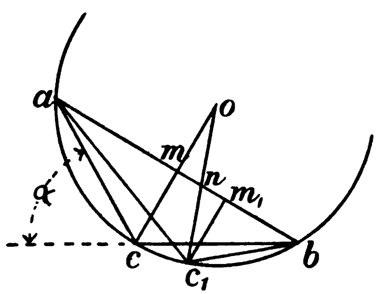


FIG. 75.

Therefore v^2 is least when cm is a maximum.

This occurs when m is the middle point of ab.

For, draw any other triangle, abc' (Fig. 75), on base ab and with angle $ac'b$ = angle acb.

Then the points a, c, c', b are on the circumference of a circle whose centre will be on cm produced.

Let o be the centre.

Join oc', cutting ab at n.

Draw $c'm'$ perpendicular to ab.

Then om is less than on.

Therefore $oc - om$ is greater than $oc' - on$.

Therefore cm is greater than $c'n$, and therefore greater than $c'm'$.

Therefore cm is a maximum when m is the middle point of ab.

Therefore v^2 is least when m is the middle point of ab; that is, when $bc = ac$, or when $bc = \dfrac{ab}{2\cos\frac{a}{2}}$.

That is, v^2 is least when

$$W = \frac{V}{2 \cos \frac{a}{2}} \text{ and } \gamma = \tfrac{1}{2} a$$

It should be noted that this holds good only when a and β are fixed, and V is fixed in magnitude; for equation (2), p. 65, shows that any alteration in the values of V, a or β will also affect the value of v^2.

Suppose now that V is fixed in direction as well as in magnitude, and that a and β are not fixed, but are equal to each other. We want to find in this case what is the best value for W and for a and β in order that v^2 may be a minimum.

Since $a = \beta$, we have from equation (2), p. 65—

$$v^2 = V^2 - 4\, bc \,.\, ac \,.\, \cos a$$
$$= V^2 - 4\, bc \,.\, cg$$

Therefore v^2 is least when $bc \,.\, cg$ is a maximum, and as bg is a constant (being equal to V cos γ), this will occur when c is the middle point of bg, that is, when $W = \frac{1}{2} V \cos \gamma$, and tan a and tan β each $= 2 \tan \gamma$.

A third case occurs when V, γ, and a are fixed. The triangle abc (Fig. 73) is then fixed, and with it the value of W. β is then the only variable on the right-hand side of equation (2), p. 65, and it therefore follows that v^2 will be a minimum when $\beta = 0$. If B cannot be made equal to zero, it ought obviously to be made as small as possible.

By way of example now suppose that a and β have been fixed at 45° each, and that V = 3400 feet per sec.

The best value for W will then be $\frac{3400}{2 \cos 22\frac{1}{2}^\circ} = 1840$ feet per sec., and the best value for γ will be $22\frac{1}{2}^\circ$.

Since *cb, cd, eb* and *ed* (Fig. 73) are all equal—

$$\text{Therefore the angle } ecb = \tfrac{1}{2} \text{ of angle } dcb$$
$$= \tfrac{1}{2}(180^\circ - 45^\circ)$$
$$= 67\tfrac{1}{2}^\circ$$

$$\text{Therefore } ec = \frac{eh}{\sin 67\frac{1}{2}^\circ} = \frac{ag}{\sin 67\frac{1}{2}^\circ} = \frac{ab \sin 22\frac{1}{2}^\circ}{\sin 67\frac{1}{2}^\circ}$$

$$\text{Therefore } ec^2 = \frac{ab^2 \sin^2 22\frac{1}{2}^\circ}{\sin^2 67\frac{1}{2}^\circ} = 0{\cdot}17ab^2$$

$$\text{Therefore } v^2 = 17 \text{ per cent. of } V^2$$

That is, 17 per cent. of the energy of the fluid is lost, and the fluid efficiency is 83 per cent.

As a rule, the angles α and β are not made equal to zero in order to allow the fluid to enter and to leave the buckets. These angles may, however, be made each equal to zero if the plane of motion of the fluid in any bucket is arranged to make an angle with the line of motion of the bucket.

To make this clear, consider that Fig. 76 is a section of a bucket by the plane in which the fluid moves while in the

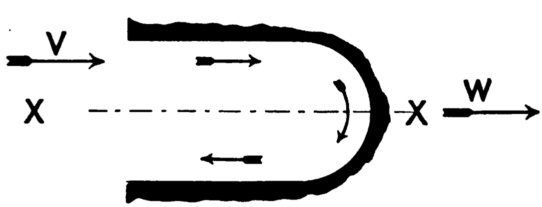


Fig. 76.

bucket (the plane is indicated by the line YY on Fig. 77). Let Fig. 77 be a section on the line XX of Fig. 76.

The arrow W indicates the direction of motion of the bucket. The arrow V indicates the jet of fluid about to enter

the bucket. The small arrows indicate the direction of flow of the fluid when in the bucket relatively to the bucket.

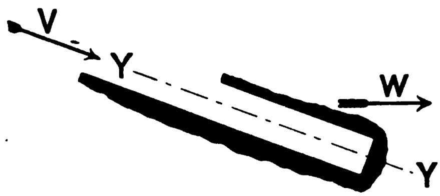


FIG. 77.

Referring now to Fig. 78, let CB represent in magnitude and direction the velocity W of the bucket, and let AB represent in magnitude and direction the absolute velocity, V, of the jet of fluid about to enter the bucket. Then AC will represent

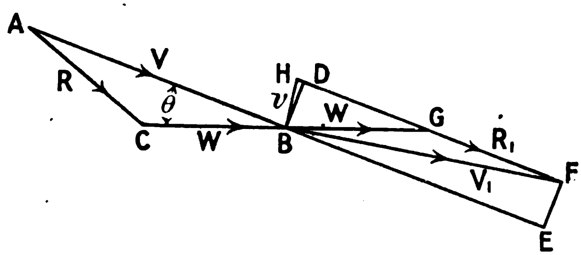


FIG. 78.

R, the velocity of the jet relatively to the buckets before it enters the buckets. This velocity R does not much concern us in this case, for when the fluid enters the buckets it has impressed on it another velocity. The velocity W can be resolved into two components, namely, W cos θ parallel to V, and W sin θ perpendicular to V. If BG represent W, then DG and BD will respectively represent the two components. The component BD will be impressed upon the jet when the latter enters the bucket, and therefore, if BE represent V, and we complete the parallelogram EBDF, the diagonal BF will

represent the absolute velocity of the jet just after it has entered the bucket. We shall call this velocity V_1 : note that it is greater than V.

As BG represents the velocity of the bucket, GF will represent the velocity of the jet relatively to the bucket just after it has entered the bucket. We shall call this R_1. The corresponding velocity r, when the fluid is leaving the bucket (if the sectional area of the stream and the density remain constant), will obviously be equal to $-R_1$, as the direction is completely reversed. Let this be represented by GH. The absolute velocity, v, with which the fluid leaves the turbine will be the resultant of r and W, and will therefore be represented by BH.

As before, it is desirable to have v^2 as small as possible.

Referring to the triangle BHG—

$$BH^2 = GH^2 + BG^2 - 2BG \, . \, GH \, . \cos \theta$$

$$i.e. \ v^2 = R_1{}^2 + W^2 - 2WR_1 \cos \theta \quad . \quad . \quad . \quad (3)$$

$$\text{but } GF = DF - DG$$

$$\text{that is, } R_1 = V - W \cos \theta.$$

Supplying this value of R_1 in equation (3), we have—

$$\begin{aligned} v^2 &= V^2 + W^2 \cos^2 \theta - 2VW \cos \theta + W^2 - 2 \, W \cos \theta(V \\ &\qquad - W \cos \theta) \\ &= V^2 + W^2 \cos^2 \theta - 2VW \cos \theta + W^2 - 2VW \cos \theta \\ &\qquad + 2W^2 \cos^2 \theta \\ &= V^2 + 3W^2 \cos^2 \theta + W^2 - 4VW \cos \theta \\ &= V^2 + W^2(3 \cos^2 \theta + 1) - 4VW \cos \theta \quad . \quad . \quad . \quad . \quad . \quad . \quad (4) \end{aligned}$$

This is a minimum when $\dfrac{d(v^2)}{dW} = 0$

That is, when $2W(3 \cos^2 \theta + 1) - 4V \cos \theta = 0$

$$\text{That is, when } W = \frac{2V \cos \theta}{3 \cos^2 \theta + 1} \quad . \quad . \quad . \quad . \quad (5)$$

When W has this value, we find from equation (4) that—

$$v^2 = V^2 + \left(\frac{2V \cos \theta}{3 \cos^2 \theta + 1}\right)^2 (3 \cos^2 \theta + 1) - \frac{4V \cos \theta \times 2V \cos \theta}{3 \cos^2 \theta + 1}$$

$$= V^2 + \frac{4V^2 \cos^2 \theta}{3 \cos^2 \theta + 1} - \frac{8V^2 \cos^2 \theta}{3 \cos^2 \theta + 1}$$

$$= V^2 - \frac{4V^2 \cos^2 \theta}{3 \cos^2 \theta + 1}$$

$$= \frac{3V^2 \cos^2 \theta + V^2 - 4V^2 \cos^2 \theta}{3 \cos^2 \theta + 1}$$

$$= \frac{V^2 - V^2 \cos^2 \theta}{3 \cos^2 \theta + 1}$$

$$= \frac{V^2(1 - \cos^2 \theta)}{3 \cos^2 \theta + 1} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

Now suppose, by way of example, that $V = 2000$ feet per sec., and that $\theta = 20°$.

Then the value of W, to give the maximum fluid efficiency, is obtained by supplying the values of V and θ in equation (5), thus—

$$W = \frac{4000 \cos 20°}{3 \cos^2 20 + 1}$$
$$= 1030 \text{ feet per sec.}$$

Supplying the values of V and θ in equation (6), we obtain—

$$v^2 = \frac{V^2(1 - \cos^2 20)}{3 \cos^2 20 + 1}$$
$$= 0{\cdot}03V^2$$

That is, 3 per cent. of the energy of the fluid is lost, and the fluid efficiency is 97 per cent.

The impact, or "shock," of the bucket on the entering jet will, however, reduce this fluid efficiency by causing heat and eddies. The production of heat in a steam turbine, although diminishing the fluid efficiency, may increase the mechanical efficiency by drying or superheating the steam, and thus reducing friction.

It sometimes happens that the best value for W has to be sacrificed to the calls of safety, cheapness of construction or convenience, or it may be that a reduction in the value of W below its best amount for fluid efficiency is justified by a consequent gain in mechanical efficiency. If the area of section of the jet or stream of fluid varies in passing through the bucket, or if its density varies, then the relative velocity of the fluid when leaving the bucket may not equal the relative velocity just after it has entered the bucket. That is, in Fig. 73, *cd* may not equal *ac*, and in Fig. 78, GH may not equal GF. This will, of course, effect the best value of W. If it is known how the relative velocity of the fluid in the buckets varies, then, from what has already been said, it will be evident how the best value of W can be found graphically by trial and error.

The relative velocity of the fluid in the buckets may vary owing to friction, but friction does not necessarily cause a variation in the velocity. The work absorbed by friction may be provided by a fall of pressure.

With hydraulic turbines V is comparatively small; 150 feet per second is a high value. In steam turbines, however, V is immensely greater.

If steam at a high pressure is allowed to escape through a

small, sharp edged orifice in a plate into the open air or into a chamber at a lower pressure, it is found that only a small portion of its heat is converted into kinetic energy. If, however, the steam is allowed to escape through a diverging nozzle, a much larger proportion of its heat energy is converted into kinetic energy, and, if the nozzle is suitably designed, the steam will expand adiabatically right down to the pressure in the vessel, or in the medium into which it is discharged.

Suppose that a pound of dry saturated steam at 285 lbs. pressure absolute is expanded adiabatically and without doing work on anything but itself through a divergent nozzle into a chamber in which the pressure is 0·6 lbs. abs., then 26·7 per cent. of the steam will be condensed, and the heat energy given up will be 382 British thermal units.

Therefore K.E. $= 382 \times 778$ foot-lbs.

Therefore, if $V =$ velocity of issuing jet in feet per second—

$$\frac{V^2}{2g} = 382 \times 778$$

$$\text{Therefore } V = \sqrt{382 \times 778 \times 2g} = 4370 \text{ feet per second.}$$

If this steam be allowed to act on a single ring of vanes in a steam turbine, then, as we saw that to obtain a good efficiency the velocity of the vanes must never be much less than half the velocity of the entering fluid, it follows that the velocity of the vanes should not be less than 2000 feet per second.

Now, it can be proved that if a ring, whose thickness measured radially is not great compared with its mean diameter, be rotated about its axis, the stress produced in the material

due to centrifugal force will be approximately wv^2; * where w is the density of the material (*i.e.* mass per unit volume), and v is the mean velocity, that is, the velocity at a point at the end of a mean radius. The width of the ring, measured parallel to its axis, and the mean radius do not affect the result. Even if the thickness of the ring, measured radially, is great compared with the mean diameter, the result is not greatly altered.

* Let the velocity at outer circumference of the ring shown in Fig. 78A be represented by V_1, and the radius to the outer circumference by R_1.

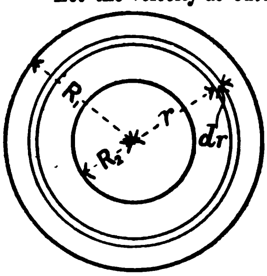


FIG. 78A.

Therefore velocity at any radius $r = \frac{V_1 \times r}{R_1}$.

Let w = density of material.

Therefore centrifugal force at radius r on ring of indefinitely small breadth dr $\Bigg\} = \frac{2\pi r \,.\, dr \,.\, w}{r} \times \frac{V_1^2 \times r^2}{R_1^2}$

$$= \frac{2\pi \,.\, dr \,.\, w \,.\, V_1^2 \,.\, r^2}{R_1^2}$$

$$\text{Therefore total C.F.} = \int_{R_2}^{R_1} \frac{2\pi dr \,.\, w \,.\, V_1^2 \,.\, r^2}{R_1^2}$$

(where R_2 = interior radius of ring A)

$$= \frac{2\pi w V_1^2}{R_1^2} \int_{R_2}^{R_1} r^2 \,.\, dr$$

$$= \frac{2\pi w V_1^2}{R_1^2} \times \frac{R_1^3 - R_2^3}{3} = \frac{2\pi w V_1^2 (R_1^3 - R_2^3)}{3R_1^2}$$

Let v = velocity at mean radius from centre.

$$\text{Then } v = \frac{R_1 + R_2}{2} \times \frac{V_1}{R_1} \text{ or } V_1 = \frac{2vR_1}{R_1 + R_2}$$

$$\text{Therefore total C.F.} = \frac{2\pi w(R_1^3 - R_2^3)}{3R_1^2} \times \frac{4v^2 R_1^2}{(R_1 + R_2)^2} = \frac{8\pi w v^2 (R_1^3 - R_2^3)}{3(R_1 + R_2)^2}$$

$$\text{Now, the force tending to break the ring across a diameter} = \frac{\text{C.F.}}{\pi}$$

$$\text{Therefore average stress of material} = \frac{\text{C.F.}}{\pi \times 2(R_1 - R_2)} = \frac{8\pi w v^2 (R_1^3 - R_2^3)}{3(R_1 + R_2)^2 \times 2\pi(R_1 - R_2)}$$

$$= \frac{4wv^2(R_1^2 + R_1 R_2 + R_2^2)}{3(R_1 + R_2)^2}$$

When R_2 approaches R_1, the stress approaches $\frac{4wv^2 \times 3R_1^2}{3 \times 4R_1^2}$, which $= wv^2$

If a steel ring, therefore, weighing 500 lbs. per cubic foot, could have a mean velocity of 2000 feet per second, the stress produced in it would be nearly 200 tons per square inch of cross-section. If the interior diameter of the ring and its velocity there be fixed, then any increase in the external diameter will increase the stress.

This shows that when high-pressure steam acts on a single set of vanes, the efficiency of a turbine using it is limited by the strength and weight of the materials available for its construction. This difficulty may be overcome by making the steam act in series on several vanes so that the velocity of these may be moderate and yet efficient. It has already been shown that the velocity of the vanes must be somewhere about half of the velocity of the entering fluid, if the velocity of the fluid when leaving the vanes is to be a minimum. But the latter velocity need not be a minimum if the fluid has to act on another set of vanes. The fluid may act on several sets of vanes in succession, and the angle and velocities of these vanes may be so arranged that the fluid gives up a portion of its energy to each.

In Fig. 79 let *ab* represent the absolute velocity of the fluid entering the first set of vanes. Let α and β be the angles of the vanes at the points of entrance and exit of the fluid, and let *cb* represent the velocity of the vanes. Then *ac* represents the velocity of the fluid relatively to the vanes as it enters, and *cd* its velocity relatively to the vanes as it leaves. If the sectional area of the fluid while passing through between the vanes is constant, and if the fluid neither expands nor contracts in volume, then $cd = ac$; *ce* will represent the absolute velocity of the fluid when it leaves the first set of vanes. If the fluid be then guided so that it takes the direction *ef*,

and if *ef* be made equal in length to *ce*, then *ef* will represent the absolute velocity of the fluid as it enters the second set of vanes. If these vanes are similar to the last, and have the same velocity which is here represented by *mf*, the relative velocity of the fluid entering the vanes will be represented by *em*, and the fluid leaving this set of vanes will have a relative velocity represented by *mn*, which is equal to *em*, and an absolute velocity represented by *mo*. If the fluid be now guided into the direction *op*, and made to act on another set of similar vanes, having a similar velocity represented by *qp*, the fluid will leave this set of vanes with an absolute velocity represented by *qs*. It will thus be seen that the energy taken from the fluid, which is proportional to $ab^2 - qs^2$, is a large proportion of the total available energy, which is proportional to ab^2; but the velocity of the vanes is only a small fraction of the initial velocity of the fluid. By having a greater number of sets of vanes, the velocity of these could be kept still lower.

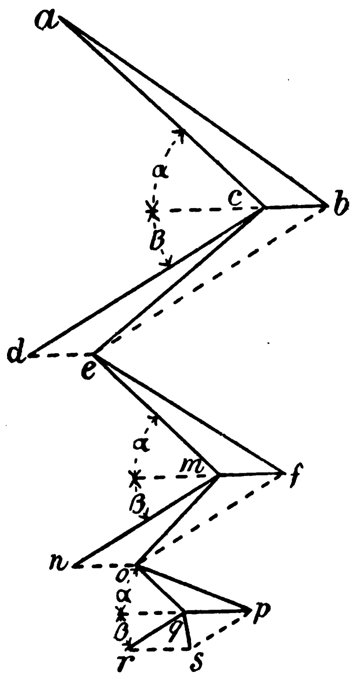


FIG. 79.—Diagram showing Velocities of Fluid in a Turbine in which the steam passes in series through several sets of vanes.

The several sets of vanes can be all arranged on the same shaft. If all the sets are placed the same distance from the axis of the shaft, *cb*, *mf*, and *qp* will be equal. Otherwise these lines will be unequal.

We have assumed that the fluid neither expands nor contracts in volume, and that its sectional area does not vary from the time it enters the first set of vanes to the instant it leaves the last set. If the fluid is steam, however, it will change its state in passing in series through several sets of vanes. Friction will in every case absorb some of its energy, and this will cause either a reduction of velocity, a drop of pressure or condensation. As a change of bulk means a change of velocity unless the section of the jet or stream is correspondingly altered, it follows that the velocity of steam passing in series through several sets of buckets will, as a rule, vary otherwise than as shown in Fig. 79. Fig. 79A is a diagram for steam passing through the same vanes moving at the same speed as in Fig. 79, but with the velocity of the steam relatively

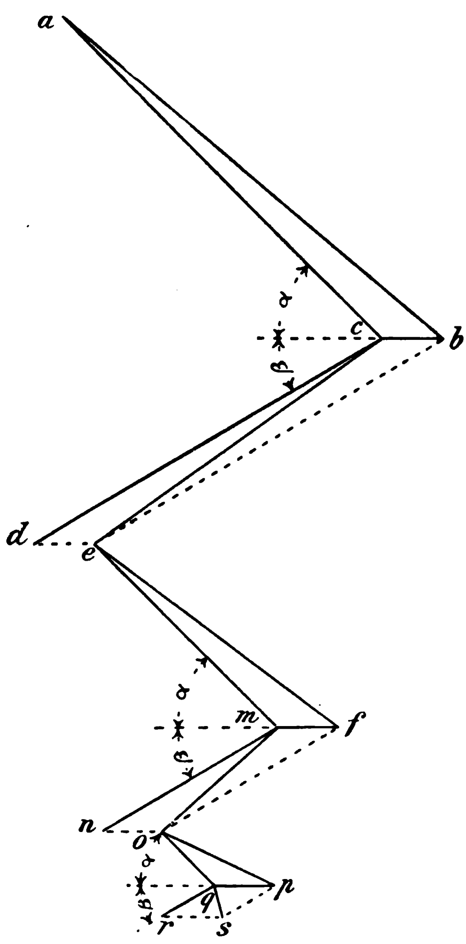


FIG. 79A.—Diagram showing Velocities of Fluid in a Turbine in which the steam passes in series through several sets of vanes.

to the vanes diminishing in passing through each set of vanes, and with the absolute velocity of the steam diminishing in passing from one set to the next. That is to say, *cd* is less than *ac*, *ef* is less than *ce*, *mn* is less than *em*, and so on.

Fig. 80 is a diagram for steam passing through the same vanes moving at the same speed as in Figs. 79 and 79A, but with the velocity of the steam relatively to the vanes increasing in passing through each set of vanes, and with the absolute velocity of the steam increasing in passing from one set to the next. *cd* is therefore greater than *ac*, *ef* than *ce*, and so on.

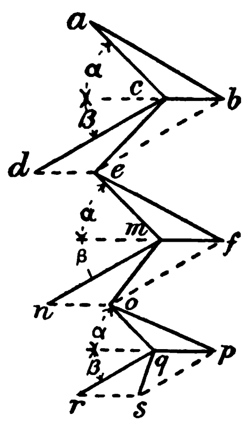


FIG. 80.—Diagram showing Velocities of Fluid in a Turbine in which the steam passes in series through several sets of vanes.

It must not be thought that these last three diagrams illustrate actual velocities that have been obtained in practice with steam turbines; the diagrams are merely intended to make clear the theory of the flow of steam through several sets of buckets in series. The velocity of steam in any part of a turbine depends on a great many conditions. The initial velocity, pressure and temperature, the angles of the vanes, the section of the passages for the flow of steam, the amount of friction, the terminal pressure and the amount of radiation, all affect the velocity of the steam at any point, so that this velocity can be given almost any value.

In a multiple-expansion steam turbine, such as the Parsons, the fluid is purposely allowed to expand in passing in series through the several rings of blades. In a Parsons steam

turbine, practically the whole of the expansion of the steam takes place after the fluid has entered the first set of vanes, and, as the steam passes through a great many sets of vanes, its velocity is never excessive, and is usually under 600 feet per second. As, moreover, with a number of sets of vanes, the velocity of the vanes need only be a small fraction of the velocity of the steam, it follows that vane speeds can be kept comparatively low without losing efficiency. Very good results have been obtained with Parsons turbines running at nearly as low a speed as that of fast reciprocating engines.

Fig. 81 shows the fixed and moving vanes or blades of a parallel-flow turbine of the Parsons type, the dotted line and small arrow-heads showing the passage of the steam. The fixed blades are for guiding the steam from one set of moving blades to the next. The relative and absolute velocities at different points are lettered to correspond with Figs. 79, 79A, and 80. The lines *ab*, *ac*, etc., are intended to represent only the directions, and not the magnitudes of the velocities. The fall of pressure of the steam in passing through a Parsons turbine is very nearly along an adiabatic line.

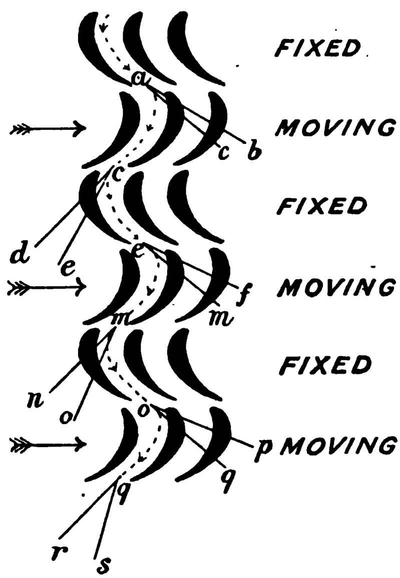


FIG. 81.—Passage of Steam through a Parsons Turbine.

By increasing the number of rings of blades in a multiple-

expansion steam turbine, the drop of pressure per ring will be reduced. The velocity of flow of the steam through each ring of blades will consequently be diminished, and a lower efficient vane speed therefore obtained. There are, however, practical objections to having too great a number of rings of blades between the journals of the turbine spindle. A spindle having a great length between supports, springs or whips, thereby causing unsteady running. This necessitates larger clearances, which usually mean greater leakage of steam past the vanes. When a low vane speed is desired, compounding is therefore desirable. The turbine is made in two or more parts, and the steam passed in series through these. The several parts of the turbine may be arranged to drive the same shaft or different shafts. If a high-pressure and low-pressure cylinder are arranged on the same shaft, the axial thrust of the one may be used to balance the axial thrust of the other.

The speed of revolution of the turbine spindle may be reduced, without reducing the vane speed, by increasing the diameter of the rings of blades. In the case of a Parsons turbine, however, this involves, for equal radial clearance, a greater leakage area. As, moreover, the blade length has to be reduced to give the same passage for steam, the ratio of the amount of steam that leaks past the rings of blades to the amount which passes through them increases rapidly when the diameter of the turbine is increased without increasing the power. The diameter of a Parsons turbine is therefore limited for any power and steam pressure.

In order to reduce the velocity of the fluid acting on the vanes of a steam turbine, it has been proposed to cause the steam jet, by an injector action, to draw in air, water, or other fluid at atmospheric pressure, the velocity of the combined fluid

being thus made moderate. This would allow of a lower efficient vane speed; but, as a greater mass of fluid would leave the turbine, and as this fluid must have a certain velocity, the energy thus lost would be increased without any increase in the energy entering the turbine.

CHAPTER VI.

ENTROPY AND ENTROPY-TEMPERATURE DIAGRAMS.

As we shall be dealing with entropy-temperature diagrams, and as this subject is not very well known, it may be advisable, in the first place, to explain what is meant by "entropy," and what can be determined by an entropy-temperature, or, as it is sometimes called, a theta-phi diagram. To an engineer accustomed only to diagrams in which the ordinates and abscissæ represent readily appreciable quantities, such as pressure, or volume, or steam consumption, the idea of entropy is rather difficult to grasp. This "ghostly quantity," as Professor Perry calls it, is not perceptible by the senses, and cannot be measured directly by any gauge or meter. It is, nevertheless, a very convenient term of expression, and entropy-temperature diagrams are very instructive and very useful.

In an ordinary pressure-volume or pressure-distance diagram, as, for example, an indicator diagram, the ordinates represent pressure, the abscissæ represent volume or distance travelled, and the areas represent energy received or rejected, or work done. Now, when heat is put into or taken out of a substance, any small part of the heat so dealt with is equal to the temperature at which it was put in, or taken out, multiplied by some quantity. This quantity is called change of entropy, or difference of entropy.

In order that the temperature may be constant while the small amount of heat is being put in or taken out, it is necessary for a general case that the small part of the heat should be indefinitely small. Suppose this to be the case.

Let ϕ represent entropy, and $d\phi$ an indefinitely small change of entropy.

Let Q represent quantity of heat put into or taken from a substance, and dQ an indefinitely small change in the quantity of heat held by the substance.

Let τ represent temperature, and $d\tau$ an indefinitely small difference of temperature.

Then by our definition $dQ = \tau \times d\phi$.

Now, the total heat supplied to or taken from the body equals the sum of all the indefinitely small parts, and therefore equals the sum of all the items $\tau \times d\phi$.

This is expressed by saying

$$Q = \int dQ = \int \tau d\phi$$

This may be expressed in the form—

$$\phi_2 - \phi_1 = \int_{\tau_1}^{\tau_2} \frac{dQ}{\tau} \quad . \; . \; . \; . \; . \; . \; . \; . \; . \; (3)$$

If, therefore, we draw a curve, ACB (Fig. 82), such that its ordinates A*a*, C*c*, B*b* (that is, the distances of points in it from OX) represent temperature, and such that the areas *a*AC*c* and *c*CB*b* enclosed between the curve, the line OX, and any two ordinates, represent quantities of heat put into or taken out of a substance, then it is clear that the distances *ac* and *cb* will represent differences of entropy, and that the abscissæ O*a*, O*c*, O*b* (or distances of points in the curve from OY) will represent the entropy of the substance at the points A, C, B.

Temperatures are reckoned from absolute zero, which is represented by the line OX. Entropy may be reckoned from any point, but it is convenient, in dealing with water and steam, to consider zero entropy to be that of water at freezing-point (32° F.). This will then be represented by OY. Quantities of heat always refer to one pound of the working substance.

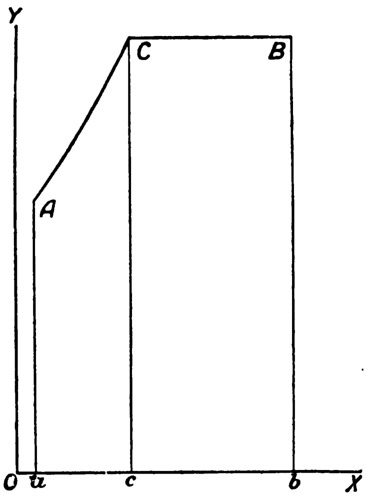


FIG. 82.—Entropy-temperature Diagram.

In Fig. 83 AB is an entropy-temperature curve for water raised in temperature from 100° F. to 400° F. The temperatures are indicated both on the ordinary Fahrenheit and on the absolute scale. It will be seen that the difference of entropy between water at 100° F. and water at 400° F. equals 0·437. The amount of heat required to effect this physical change in the water is represented by the area $aABb$. The curve AB

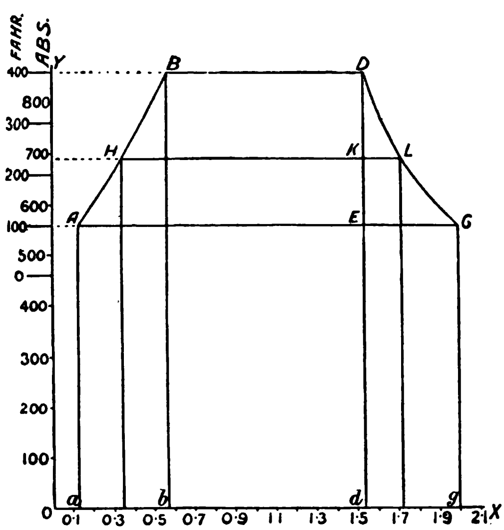


FIG. 83.—Entropy-temperature Diagram for Water and Steam.

may be drawn by obtaining from a table the entropy of water at several temperatures, and plotting these values. If the water at 400° F. be converted into steam at that temperature, the entropy-temperature curve will be parallel to OX, as the temperature is unchanged. The change of entropy will be represented by *bd*, and the heat put into the substance by the area BD*db*. The heat put into the water is obviously the latent heat of steam at 400° F. (860° abs.). This equals 830 units. The change of entropy *bd* is therefore equal to $\frac{830}{860}$, or 0·965, as indicated on the diagram. If, now, the steam expand adiabatically against a resistance, the temperature will fall, but as no heat is being imparted to or taken from the steam, it is obvious that the area below the curve of expansion must be zero—that is, that no change of entropy will take place. The entropy-temperature curve will therefore lie along D*d*, and will be represented by DE if the temperature fall to 100° F. If heat be now abstracted from the steam and water (for some of the steam will have condensed during expansion) till all the fluid exists in the liquid state, but without lowering the temperature, the entropy-temperature curve will be EA, which is parallel to OX. The quantity of heat taken from the fluid will be represented by the area *a*AE*d*. The total heat supplied to the fluid is therefore proportional to the area *a*ABD*d*, and the heat abstracted to the area *a*AE*d*. The heat converted into work is therefore proportional to the area ABDE, and the efficiency of a heat-engine working on this cycle $= \dfrac{\text{area ABDE}}{\text{area } a\text{ABD}d}$.

If, instead of allowing the steam to expand adiabatically, we had, during expansion, supplied just sufficient heat to it to maintain it in a dry, saturated condition, the entropy-temperature curve for expansion would be DG instead of DE. If heat had

then been taken from the steam without reducing its temperature till the whole had condensed, the drop of entropy would be represented by AG (or *ag*), and the quantity of heat abstracted by the area *a*AG*g*. This last quantity is obviously the latent heat of steam at 100° F., and it is evident that the ratio of the area *a*AE*d* to the area *a*AG*g* is the fraction of the latent heat available to be given up after the steam has expanded, according to the line DE. This ratio must therefore represent the fraction of the steam uncondensed at E. The areas are proportional to the lines AE and AG, and therefore $\frac{\mathrm{AE}}{\mathrm{AG}}$ represents the dryness fraction, or the fraction of the steam uncondensed after the adiabatic expansion DE has taken place, or when the point E is reached during the isothermal withdrawal of heat GA. Similarly, if any other horizontal line such as HKL be drawn, $\frac{\mathrm{HK}}{\mathrm{HL}}$ will represent the dryness fraction of the steam at the point K of the adiabatic expansion DE.

The curve DG may be drawn by obtaining from a table the entropy of dry, saturated steam at several temperatures, or it may be obtained in another manner. AG × A*a* = area *a*AG*g*. But *a*A represents a certain temperature, and *a*AG*g* represents the latent heat of steam at that temperature. Therefore the length of AG can be obtained by dividing the latent heat by the temperature. Several horizontal lines, such as AG and HL, can thus be determined, and the curve DLG drawn through their ends.

CHAPTER VII.

THEORETICAL CONSIDERATION OF DIFFERENT TREATMENTS OF STEAM IN A HEAT-ENGINE.

It is intended in this chapter to consider the effects of treating steam in different ways on the efficiencies of heat-engines with special reference to the steam turbine.

Let us consider the transfer of heat energy into mechanical energy in a heat-engine or apparatus comprising a boiler in which water is heated to a certain temperature and then converted into steam, a turbine or other motor in which the steam is expanded and loses some of its heat, and a condenser in which more of the heat is taken from the fluid before the latter is returned to the boiler.

The different cases which will be considered have been chosen not to represent what occurs in practice, but to indicate the effects of different treatments of the steam, so that it can be ascertained what had best be done with any type of turbine, in order to prevent waste and promote efficiency, and what is likely to be gained or lost by any alteration in treatment, such, for example, as by superheating the steam.

Case I.

Let us suppose, in the first instance, that feed water is received into a boiler at 85° F., and heated to 382° F. The

entropy-temperature curve for this heating is shown at AB, in Fig. 84.

Suppose that the water is converted into steam at this temperature, which means that the pressure is 200 lbs. absolute. This absorption of heat is represented by BC. Let the steam now expand adiabatically, doing work, till the pressure falls to 0·6 lb. absolute. The temperature corresponding to this pressure is 85° F. This expansion is represented by the line CD on the diagram. Some of the steam will condense during this expansion, and we can find the wetness at any point in CD, by the method described in connection with Fig. 83. Lastly, let heat be abstracted from the fluid till the whole of the steam has condensed, but without any reduction of temperature, and let the water be returned to the boiler. This action is represented by DA on the diagram (Fig. 84). It does not matter whether the heat be abstracted from the steam in the turbine or in a condenser, or in any other vessel, provided that it takes place after the expansion and the fall in temperature are completed.

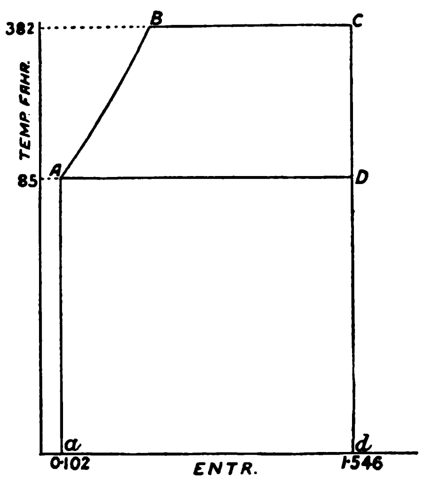


Fig. 84.—Case I.: Adiabatic Expansion; isothermal compression; range of temperature, 85° F.—382° F.

The heat supplied to the fluid is then represented by the area aABCd, and the heat abstracted by the area aADd. The heat converted into work is therefore represented by the area ABCD and—

$$\text{The efficiency} = \frac{\text{area ABCD}}{\text{area } a\text{ABC}d} = 0{\cdot}31$$

Case II.

Let us suppose now that the steam generated at 200 lbs. pressure, instead of expanding adiabatically, be supplied during expansion with sufficient heat to prevent any condensation. This might be approximately attained by jacketing a steam turbine with high-temperature steam. The condensation will then all take place at constant temperature, as shown by EA. The entropy - temperature diagram will then be as shown in Fig. 85.

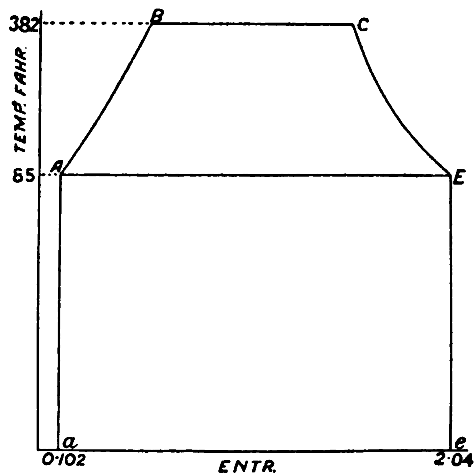


Fig. 85.—Case II.: Expansion along Line of Dry Saturated Steam; isothermal compression; range of temperature, 85° F.—382° F.

The heat supplied to the fluid is represented by the area aABCEe, and the heat abstracted by the area aAEe. The heat converted into work is therefore represented by the area ABCE, and—

$$\text{The efficiency} = \frac{\text{area ABCE}}{\text{area } a\text{ABCE}e} = 0{\cdot}28$$

Compared with Case I. it will be seen that there is an increase both in the heat supplied and in that converted into work, but the latter is not increased proportionately to the former, and hence the drop in the efficiency.

Case III.

Suppose in this case that the steam, instead of receiving heat while expanding, has heat taken from it, as, for example, by radiation from the exterior of a turbine. Suppose that by this means twice as much steam is condensed during expansion as in Case I. The entropy-temperature diagram will then be as shown in Fig. 86, where $\frac{FE}{AE}$ is the fraction of the steam that is condensed at the end of the expansion. The remainder of the steam is condensed at constant temperature, as shown by the line FA. The heat supplied to the fluid is represented by the area aABCd, and the heat withdrawn by the area aAFCd. The heat converted into work is therefore represented by the area ABCF, and—

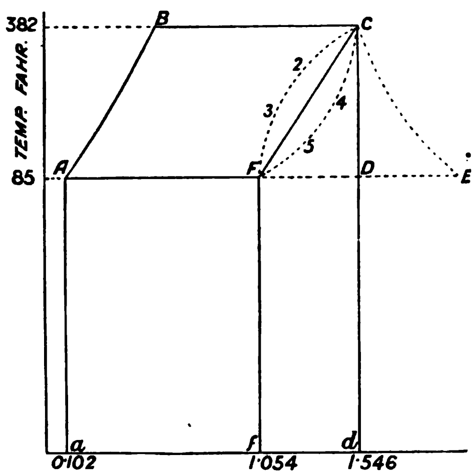


Fig. 86.—Case III.: Expansion with Leakage of heat; isothermal compression; range of temperature, 85° F. – 382° F.

$$\text{The efficiency} = \frac{\text{area ABCF}}{\text{area } a\text{ABC}d} = 0{\cdot}25$$

The heat supplied is the same as in Case I., but the portion of this that is converted into work is less than in Case I. by the amount represented by the triangle CFD. The efficiency is therefore less than in Case I.

The total leakage of heat during expansion is represented

by the area *f*FCD*d*. Of this the area CFD goes to reduce the heat converted into work, and the area *f*FD*d* to reduce the work of the condenser. There is no reason why CF should be a straight line. A straight line has only been assumed for convenience. The nature of the line would depend on the construction of the engine. (In a reciprocating engine the line is usually very convex to the left, on account of the coolness of the cylinder at the beginning of the stroke.) It should therefore be noted that if the leakage of heat had been greater than that represented towards the beginning of the expansion, and less towards the end, so that the line of expansion was as shown by the dotted line C23F, the efficiency would have been reduced; while if, on the contrary, the leakage had been less towards the beginning of the expansion and greater towards the end, as indicated by the dotted line C45F, the efficiency would have been increased; the total amount of steam condensed being the same in all cases.

Case IV.

Suppose in this case the steam is superheated before expansion takes place, from 382° F. to 540°, and that the expansion is then adiabatic. Fig. 87 is the diagram for this case,

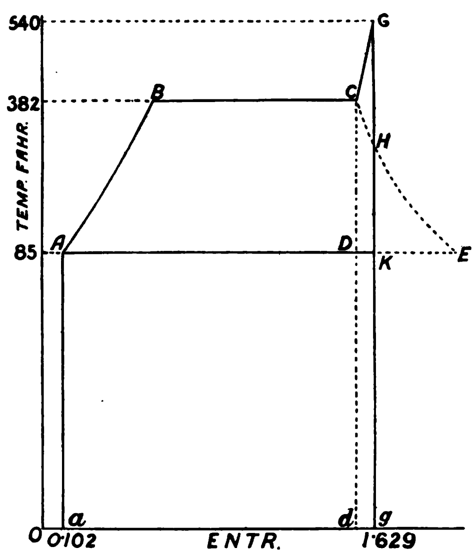


Fig. 87.—Case IV.: Superheating; Adiabatic Expansion; isothermal compression; range of temperature, 85° F.—540° F.

CG being the curve for the superheating action. It will be seen that the line GK of adiabatic expansion cuts the line CE of dry, saturated steam at the point H. This indicates to us that, during the fraction of the adiabatic expansion represented by GH, the steam is superheated; while, during the remaining fraction represented by HK, it is wet—at the point H it is dry and saturated. The heat supplied to the fluid is represented by the area *a*ABCG*g*, and the heat abstracted by the area *a*AK*g*. The heat converted into work is therefore represented by the area ABCGK, and—

$$\text{The efficiency} = \frac{\text{area ABCGK}}{\text{area } a\text{ABCG}g} = 0{\cdot}32$$

The heat supplied to the fluid during the superheating action is represented by the area *d*CG*g*. Of this the portion represented by the area DCGK is converted into work. The fraction of the heat supplied which is converted into work is therefore greater during this action than during the actions of heating the feed water and generating the steam, and it is this which raises the efficiency slightly above that in Case I.

To draw the curve CG we must make an assumption regarding the specific heat of steam at constant pressure. Let this specific heat be denoted by K, and let us assume that K is a constant, and equal to 0·48. Then from equation (3), p. 83—

$$\phi_2 - \phi_1 = \int_{842}^{1000} \frac{dQ}{\tau}$$

the numbers 1000 and 842 denoting the temperature on the absolute scale, and ϕ_1 and ϕ_2, denoting the entropy respectively before and after the superheating action.

Now $dQ = \text{K}d\tau$.

$$\text{Therefore } \phi_2 - \phi_1 = \int_{842}^{1000} \frac{K d\tau}{\tau} = 0{\cdot}48 \int_{842}^{1000} \frac{d\tau}{\tau}$$

$$= 0{\cdot}48(\log_e 1000 - \log_e 842) = 0{\cdot}48 \times 0{\cdot}1720$$

$$= 0{\cdot}08256$$

This is the difference of entropy between C and G, and determines the length *dg*. The height *g*G is of course determined by the temperature, namely, 540°. Any other point on the curve CG can be similarly located, and the curve thus obtained.

Case IVa.

In the case just described the higher limit of temperature and the range of temperature exceed that in the other cases, and therefore, in order to make a fair comparison, we must consider the case of an engine working on a cycle, as in Case IV., but with the same limits of temperature as in Cases I., II., and III.

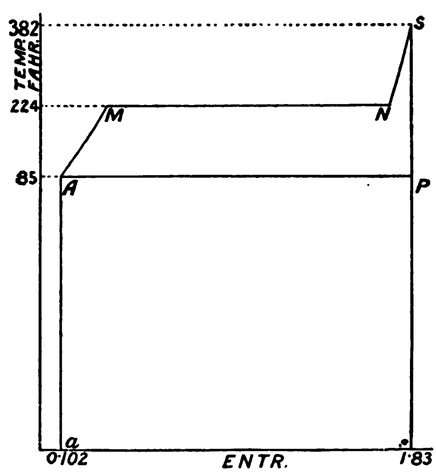


Fig. 88.—Case IVa.: Superheating; Adiabatic Expansion; isothermal compression; range of temperature, 85° F.—382° F.

Let us suppose, then, that steam is generated at 224° F., and superheated to 382° F., the cycle otherwise being the same as in Case IV. The entropy-temperature diagram will then be as shown in Fig. 88, where the heat supplied to the fluid is represented by the area

aAMNSs, and the heat withdrawn by the area aAPs. The heat converted into work is therefore represented by the area AMNSP, and—

$$\text{The efficiency} = \frac{\text{area AMNSP}}{\text{area } a\text{AMNS}s} = 0{\cdot}20$$

The efficiency is less than in Case I., because, although the maximum and minimum temperatures are the same as in Case I., most of the heat is absorbed by the fluid when at a lower temperature.

Case V.

If in Case IV. (Fig. 87), the fluid, instead of expanding adiabatically, had had the same amount of heat abstracted from it during expansion as in Case III. (Fig. 86), the entropy-temperature diagram would be as shown in Fig. 89, where the total leakage of heat is represented by the area qQJGg, which corresponds to and equals the area fFCDd in Fig. 86. The heat supplied, the heat withdrawn, and the heat converted into work are represented respectively by the areas aABCGg, aAQJGg, and ABCGJQ.

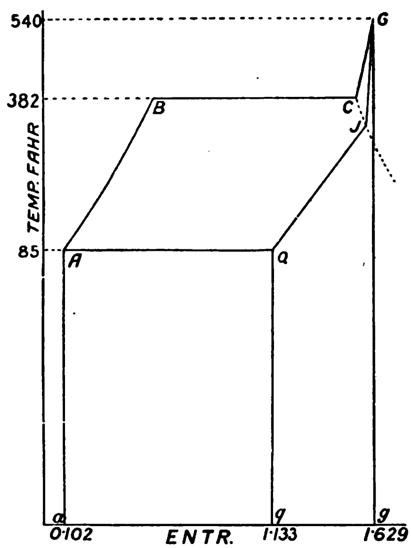


Fig. 89.—Case V.: Superheating; Expansion with Leakage of Heat; isothermal compression; range of temperature, 85° F.—540° F.

$$\text{The efficiency} = \frac{\text{area ABCGJQ}}{\text{area } a\text{ABCG}g} = 0{\cdot}26$$

The increase in efficiency in this case over Case III. is about the same as the increase in Case IV. over Case I., and for the same reason.

Case VI.

In the cases heretofore considered the steam has expanded doing work. But the steam may expand without doing work, or against an imperfect resistance. Joule found by experiment that when a gas expands without doing work, its temperature remains constant. Joule's experiment consisted in placing in a tank of water two vessels, one containing a gas under pressure, and the other empty. On communication being established between the vessels, some of the gas rushed from one vessel to the other, and the pressure fell; but it was found, after equilibrium had been established, that the temperature was the same as at the beginning of the experiment.

The phenomenon of unresisted expansion occurs when steam is passed through a reducing-valve, when the pressure falls and the steam expands without any appreciable amount of work being done. Imperfect resistance to expansion also occurs when steam passes at a high velocity through a restricted opening, and is well known in such a case by the name of "wire-drawing." When unresisted or imperfectly resisted expansion takes place, some of the heat of the gas is converted into kinetic energy; but if the gas has its velocity arrested, the energy returns to the form of heat. Thus, in the case of a reducing-valve, when the valve opens, there is a rush of steam through it, some of the heat energy of the steam being converted into kinetic energy. The rush is, however, arrested at the other side of the valve, and the kinetic energy is returned by impact or eddies to the form of heat.

In Fig. 90 AB represents the heating of the feed water, BC the generation of steam, and CX the adiabatic expansion of the steam, as in the previous cases. Suppose that free expansion then takes place till the steam is completely dried and is superheated. The state of the steam will then be represented by the point Y on the diagram. If the steam had been dried, but not superheated, the point Y would have been on the curve CZ. We cannot connect X and Y by a straight line to represent the free expansion, as this would indicate that heat had been absorbed by the steam,

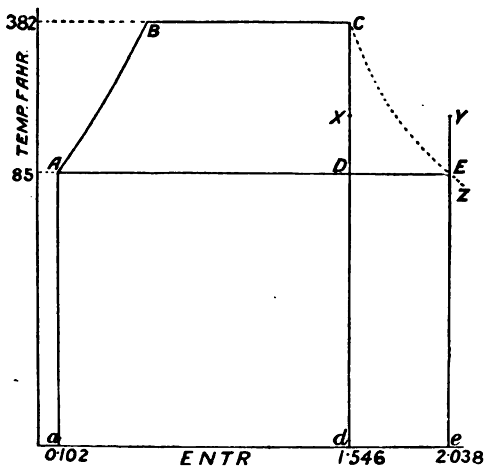


FIG. 90.—Case VI.: Expansion, partly adiabatic and partly unresisted; isothermal compression; range of temperature, 85° F.—382° F.

which is not the case. If we want an unbroken curve, we must connect X and Y by means of the straight lines X*d*, *de*, and *e*Y. Y*e* of course equals X*d*, as the temperature is unchanged. Suppose that, after the free expansion, the steam expands adiabatically, doing work, till the temperature falls to 85° F. This expansion is represented in the diagram by YE. The isothermal compression EA completes the diagram, as in the previous cases. The amount of free expansion has been so chosen that, after the last expansion, the steam is dry and saturated, as indicated, by the point E being on the curve CZ.

The heat absorbed by the fluid in this case is represented by the area *a*ABC*d* and the heat rejected by the area *a*AE*e*.

The heat converted into work is represented by the area aABCd—the area aAEe and

$$\text{The efficiency} = \frac{\text{area } a\text{ABC}d - \text{area } a\text{AE}e}{\text{area } a\text{ABC}d} = 0{\cdot}08$$

Case VII.

In this case let us suppose that the feed water is heated, the steam generated, and the adiabatic expansion commenced as in Case I.; but let the adiabatic expansion continue only till the temperature falls to 250° F., as indicated by the point V. Then let heat be abstracted from the fluid *at constant volume* till temperature 85° F. is reached, as indicated by U, Fig. 91. The isothermal compression UA completes the diagram.

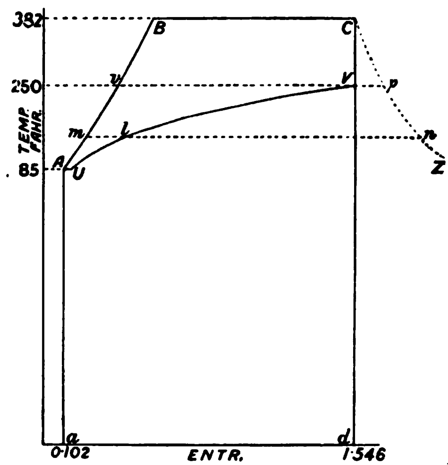


Fig. 91.—Case VII.: Adiabatic Expansion heat rejected at constant volume, followed by isothermal compression; range of temperature, 85° F.—382° F.

The curve VU is obtained as follows. Let a' be the volume of 1 lb. of saturated steam at 250° F. (the temperature at V), and let a be the volume of the same at any other temperature, τ, between V and U. Let q' be the dryness fraction of the steam at V, and q be the dryness fraction at the temperature τ. Then, neglecting the volume of the water—

$$qa = q'a'$$

because the fluid is expanding at constant volume. If l is the

required point on the curve where the temperature is τ, and mln is a horizontal line drawn to meet the line of saturated steam CZ—

$$q = \frac{ml}{mn}$$

$$\text{therefore } ml = q \times mn = \frac{q'a'}{a}mn$$

q' of course equals $\frac{v\mathrm{V}}{vp}$ (Fig. 91), and a' and a can be obtained from a table of the properties of saturated steam. Hence ml can be obtained. Similarly, any number of other points can be obtained on the curve VU.

The heat supplied to the fluid in this case is represented by the area aABCd, and the heat rejected by the area aAUVd. The heat converted into work is represented by the area ABCVU, and

$$\text{The efficiency} = \frac{\text{area ABCVU}}{\text{area } a\text{ABC}d} = 0{\cdot}18$$

This treatment, by which part of the heat is rejected at constant volume, and part at constant temperature, gives a reduced efficiency compared with the treatment in Case I., where all the heat rejected was given up at constant temperature. In reciprocating condensing engines the heat is commonly rejected, neither on a constant volume line nor on a constant temperature line, but on a line between the two. The nature of the rejection of heat in a steam turbine is pretty much a matter of conjecture.

Case VIII.

Suppose in this case that the feed water is heated, and the steam generated, superheated, and expanded adiabatically, as in Case IV., till the point T (Fig. 92) is reached, where the

temperature is 250° F. Let the fluid now expand at constant volume, as in Case VII., till the point W is reached, when the temperature is the same as at A, and let the cycle be completed by the isothermal compression WA. In this case the heat supplied to the fluid is represented by the area

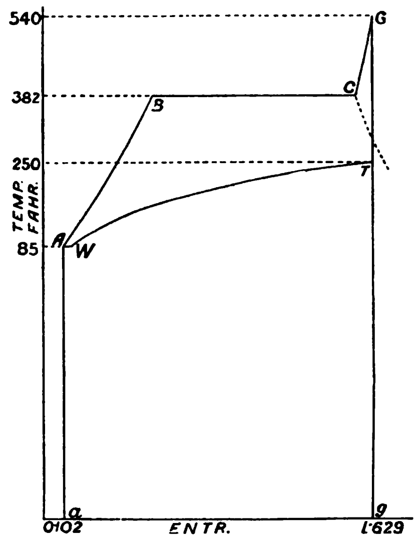


FIG. 92.—Case VIII.: Superheating; Adiabatic Expansion; heat rejected at constant volume, followed by isothermal compression; range of temperature, 85° F.—540° F.

aABCGg, as in Case IV., and the heat rejected is represented by the area aAWTg. The heat converted into work is represented by the area ABCGTW, and

$$\text{The efficiency} = \frac{\text{area ABCGTW}}{\text{area } a\text{ABCG}g} = 0{\cdot}19$$

Many more cases might be studied, but sufficient have been considered to show the effect of different treatments of the steam. The results are here tabulated in Table I.

TABLE II.

Case.	Method of treatment.	Max. temp. F.	Min. temp. F.	Efficiency.
I.	Adiabatic expansion. Isothermal compression	382	85	0·31
II.	Expansion along line of dry saturated steam. Isothermal compression	382	85	0·28
III.	Expansion with leakage of heat. Isothermal compression	382	85	0·25
IV.	Superheating. Adiabatic expansion. Isothermal compression	540	85	0·32
IVA.	Superheating. Adiabatic expansion. Isothermal compression	382	85	0·20
V.	Superheating. Expansion with leakage of heat. Isothermal compression	540	85	0·26
VI.	Expansion, partly adiabatic and partly unresisted. Isothermal compression	382	85	0·08
VII.	Adiabatic expansion. Heat rejected at constant volume, followed by isothermal compression	382	85	0·18
VIII.	Superheating. Adiabatic expansion. Heat rejected at constant volume, followed by isothermal compression	540	85	0·19

It should be borne in mind, however, that a change in the range of temperature will alter the relative efficiencies. It should also be remembered that arbitrary quantities have, as a rule, been chosen for the amount of superheating, amount of free expansion, etc.; and that, if these are altered, the results may be considerably modified. And it must not, above all things, be forgotten that there are practical considerations which affect the efficiency. For example, there is the fluid friction in a turbine. It is probable that the diminution of this fluid friction by superheating the steam accounts in great part for the increased economy obtained by superheating; for the results obtained by the tests of turbines show a greater percentage increase in efficiency with superheating than is due to thermo-dynamic reasons. Table II. shows the effect of superheating on the steam consumption of a Parsons turbine.

TABLE III.

TEST OF 500-KILOWATT TURBO-ALTERNATOR CONSTRUCTED BY MESSRS. C. A. PARSONS AND CO. FOR THE CORPORATION OF BLACKPOOL.

Pressure of steam above atmosphere at stop-valve.	Superheat at stop-valve.	Vacuum in the turbine cylinder. (Bar. = 30".)	Revolutions per minute.	Load.	Steam used.	
lbs. per sq. in.	degrees F.	ins of mercury.		kilowatts.	lbs. per hr.	lbs. per kw. hr.
146	70	27·1	2500	515	11,000	21·35
150	0	27·0	2500	502	11,600	23·1
135	0	27·3	2500	497	11,953	24·0
133	66	27·3	2500	507	10,693	21·1

The effect of superheating the steam on the efficiency of a De Laval steam turbine is well exemplified in Table XV., Chap. VIII.

CHAPTER VIII.

THE DE LAVAL STEAM TURBINE.

About 1882, Dr. Gustaf de Laval invented a turbine on the principle of Hero's engine. This turbine is illustrated diagrammatically in Figs. 93 and 94. Steam (or other fluid) entered the casing *a* by the nozzle *b*, and passed along the curved hollow arms *c, c*. These arms were formed like the buckets of an outward-flow hydraulic turbine, and the passage of the steam along them caused them to revolve and to rotate the shaft *d*. This shaft drove another shaft at a slower speed by means of friction wheels. The requisite pressure between the surfaces of these wheels was obtained by utilizing the axial thrust of the turbine wheel. The turbine shaft *d* was supported in bearings which allowed it an axial movement. This shaft (see Fig. 95) carried a bevel friction wheel *e*, and the axial thrust of the turbine wheel forced this bevel wheel against the bevel wheel *f* carried by the power shaft *g*.

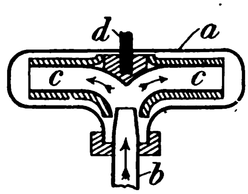


Fig. 93.

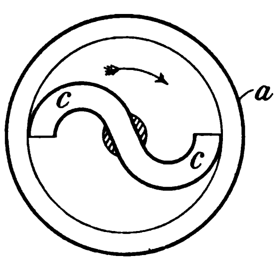


Fig. 94.
Early Turbine of Dr. De Laval's.

In 1889 Dr. De Laval applied for a British patent* for

* No. 7143 of 1889.

a steam turbine wheel combined with a diverging nozzle for the steam supply. The nozzle shown in the specification of this patent was shaped as illustrated in Fig. 96. The steam expands in passing from the smaller section *m* to the larger section *n*, and its velocity increases while its pressure falls. The object is of course to obtain a great kinetic energy with which to act on

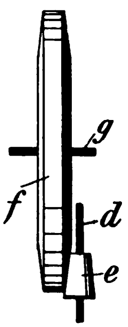


Fig. 95.—Friction Gearing.

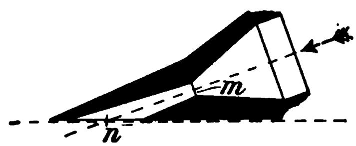


Fig. 96.—De Laval Nozzle.

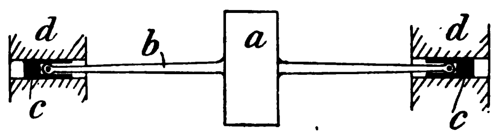


Fig. 97.—Flexible Shaft Support.

the turbine vanes. The manner in which the steam is directed on to the vanes can be seen by referring back to Figs. 2 and 2A.

Another patent of De Laval's of the same year,* refers to the flexible support of steam turbines or other bodies intended to rotate at high velocities. Figs. 97 to 106 illustrate diagrammatically several devices covered by the patent for allowing a certain amount of lateral movement to the rotating mass, to enable it to compensate for slight want of balance.

In Fig. 97, the rotating body *a* is carried on a flexible shaft, *b*, whose ends are placed in the shoes *c*, *c*, which rotate in the bearings *d*, *d*.

In Fig. 98, the rotating body *a* is flexibly connected to the shaft *b*, by providing the latter with a flange, *e*, and inserting

* No. 12,509 of 1889.

rubber rings *f*, *f*, as shown. The body is of course also supported by another shaft at the other side.

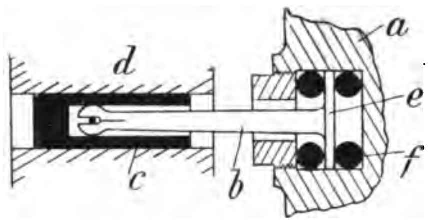


FIG. 98.—Flexibility given by Rubber Rings.

In Fig. 99, spiral springs *g*, *g*, are substituted for the rubber rings.

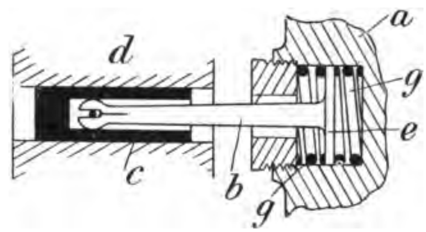


FIG. 99.—Flexibility given by Spring.

In Fig. 100, the shaft *b* is connected to the rotating body by

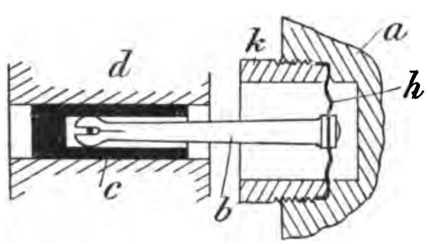
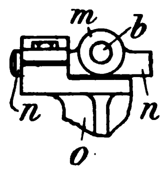


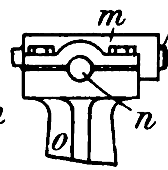
FIG. 100.—Flexibility given by Diaphragm.

means of the flexible diaphragm *h*, held in place by the gland *k*.

In the device shown by Figs. 101, 102, 103, in end elevation, side elevation, and section respectively, the shaft *b* is

supported at each end in bushes *m*, which, by means of the transverse pins *n*, *n*, can swing in the standards *o*.





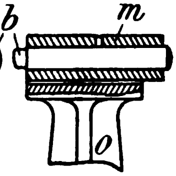
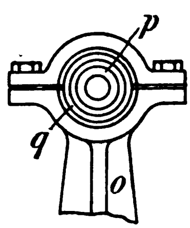


FIG. 101. FIG. 102. Fig. 103.

Flexibility given by Transverse Pivots.

In Figs. 104 and 105, the bearing bush *p* (one of these is



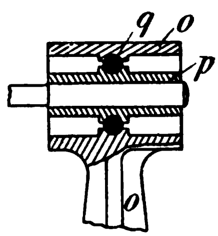


FIG. 104. Fig. 105.

Flexibility given by Rubber Ring.

provided at each end of the shaft) is supported in the cylindrical top of the standard *o*, by means of the rubber ring *q*.

In Fig. 106 the shaft is provided with spherical end pieces, *r*.

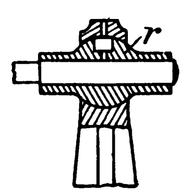


FIG. 106.—Flexibility given by Spherical End Pieces.

British patent, No. 20,603 of 1891, granted to Dr. De Laval, has reference to the exhaust passage from the turbine, which is constructed of a divergent shape in order to produce an ejector action. The velocity of the fluid at the outer end of the nozzle is less than at the inner end, owing to the increase in the section of the passage, and consequently the pressure at the inner end is less than at the

outer end. If, therefore, the pressure at the outer end is atmospheric, a partial vacuum will exist at the inner end of the passage and around the wheel, thus diminishing friction.

The form of De Laval turbine shown in Figs. 93 and 94 was intended chiefly for the direct driving of milk-separators, and milk-separators so driven are now at work.

The modern form of De Laval steam turbine was introduced about 1889. Such a machine is illustrated in Fig. 107, which shows a De Laval turbine-dynamo, as constructed

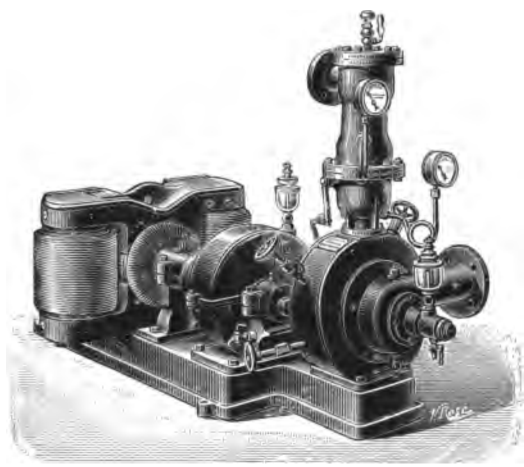


FIG. 107.—De Laval Turbine-dynamo.

by the Société de Laval (France), for horse powers from 5 to 30. The cylinder to the right contains the turbine wheel, and the intermediate cylinder is the gear box in which the high rotary motion of the wheel is geared down to a speed suitable for driving the dynamo, which is shown at the left of the figure.

Fig. 108 shows the principal parts of a turbine such as that shown in Fig. 107, but fitted with a pulley instead of being connected with a dynamo. A is the turbine shaft on which is

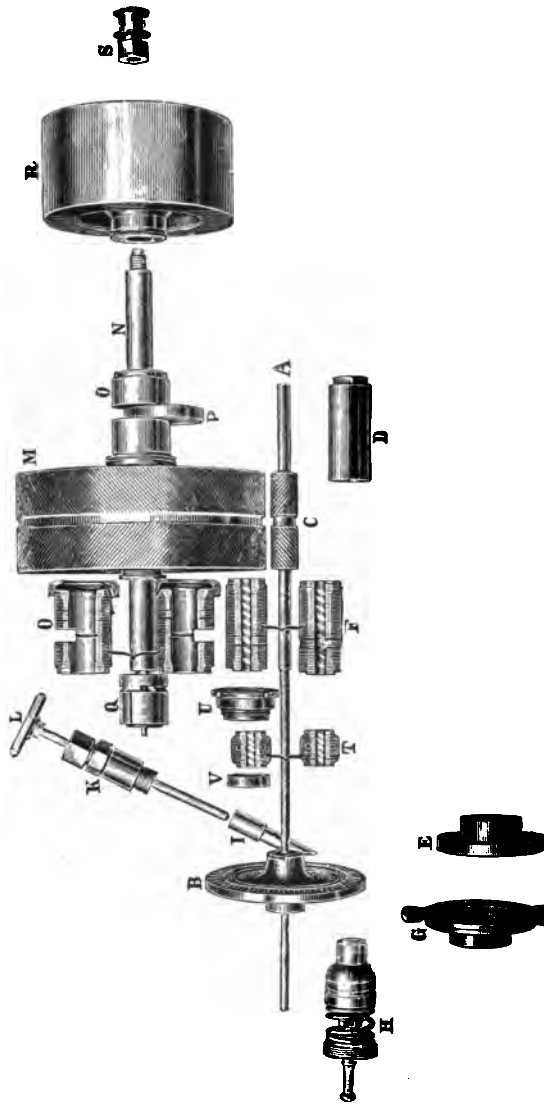


FIG. 108.—Component Parts of De Laval Turbine.

(The block for this has been prepared from an illustration kindly supplied by Messrs. Greenwood and Batley, of Leeds.)

mounted the disc or wheel B, furnished with a series of vanes. These vanes can also be seen in Fig. 109, where they are lettered W. C is a double helical pinion which gears with the toothed wheel M, the teeth on the wheel and pinion being formed at an angle of 45°, as is shown in the figure. D is the end bush of the turbine shaft, and F the middle bush, made in two parts. T is a tightening bush, also made in two parts. O, O, are the gear-wheel shaft bushes which support the power

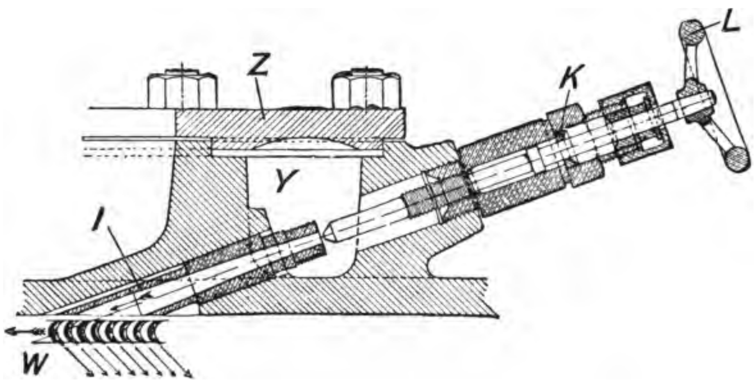


FIG. 109.—Nozzle and Vanes of a De Laval Turbine.

shaft N, which carries the gear wheel M, and the driving pulley R. S is a stop nut for the power shaft, and H a ball bush with adjusting spring for the turbine shaft. U is an adjusting nut, and V a friction gland. I is a steam nozzle, of which several are usually provided, distributed round the wheel. K is the stuffing-box for the spindle stop-valve, which can be actuated by the hand-wheel L. P is a lubricating ring, and Q is the governor which is mounted on the power shaft.

Figs. 110 and 111 show a 30-H.P. De Laval turbine dynamo in sectional elevation and sectional plan respectively. A is the turbine wheel mounted on its flexible shaft, which is journalled at S, T, Q, and R, and which carries the pinion J

gearing with the wheel K. The latter is carried on the shaft LL, which has two bearings, and is coupled at M to the armature shaft of the dynamo, which is seen at the left of the figures. The steam inlet is seen at the top of Fig. 110, with a pressure-gauge above it. The steam, after acting on the wheel, passes into a chamber G, and out through the exhaust port H.

The characteristic feature of the De Laval steam turbine is the fact that the steam is expanded its full extent before it reaches the wheel, where it arrives with its whole available energy in the form of kinetic energy. This expansion is accomplished by means of the divergent nozzles, the action being as explained on p. 73. Fig. 109 shows a common method of arranging these nozzles. It will be seen from this figure that any nozzle may be closed by screwing down the spindle, and thereby preventing the entry of steam into the nozzle from the distribution conduit Y. This distribution conduit is cast in one of the parts of the casing in which works the turbine wheel, the conduit being closed by a ring, Z. The conduit is lettered D in Fig. 111, and the steam is admitted to it after passing through a throttle or governor-valve. Several nozzles are provided for each turbine, the larger machines usually having the greater number of nozzles. As many as fifteen nozzles are sometimes provided on a large turbine. Messrs. Greenwood and Batley, Ltd., of Leeds, provide room for nine nozzles on a 225 B.H.P. turbine. All of these are employed at full load when the steam pressure is 50 lbs. When the turbine is designed for working with a higher pressure of steam, fewer nozzles are required, and the surplus holes are consequently plugged up or blind flanges put on.

The form of the nozzle I is most important. The sectional area of the smaller end has to be large enough to allow of the passage of the requisite amount of steam, while a sufficient

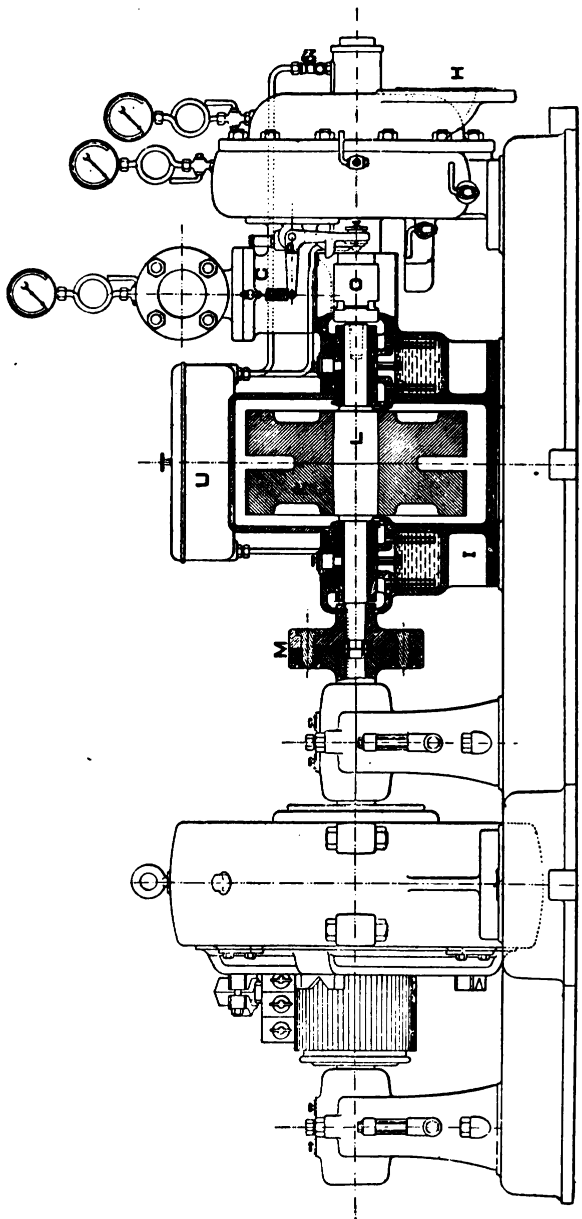


FIG. 110.—Sectional Elevation of 30-H.P. De Laval Steam Turbine Dynamo.

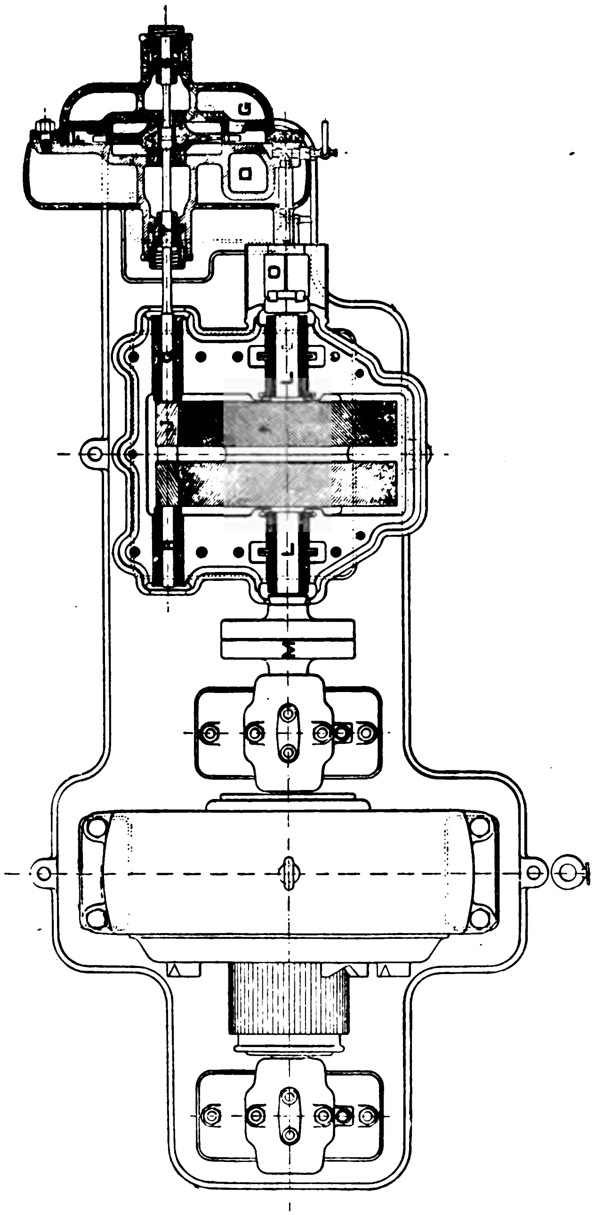


FIG. 111.—Sectional Plan of 30-H.P. De Laval Steam Turbine Dynamo.

section is required at the larger end for the complete expansion of the steam. The length of the nozzle must also exceed a certain amount, or the steam will take an eddying or irregular course through it. Too long a nozzle is objectionable, on account of friction.

For convenience of manufacture the nozzle is usually bored with a small cylindrical part at the wide end, and the rest of the nozzle with a uniform taper. If the nozzle could be as readily bored with a slightly varying taper, this would probably be done.

It was shown on page 73 that if a pound of dry saturated steam at 285 lbs. abs. be expanded adiabatically to a pressure of 0·6 lbs. abs., and have its heat energy converted wholly into kinetic energy, it will acquire a velocity of 4370 feet per second. Its kinetic energy will be 297,200 foot-lbs.

TABLE IV.

THE VELOCITY OF OUTFLOW AND THE WORKING CAPACITY OF DRY SATURATED STEAM.

Initial steam pressure, lbs. per square inch.	Counter-pressure 1 atm.			Counter-pressure 2·4 lbs. per sq. in. absolute, corresponding to 25 in. vacuum.			Counter-pressure 0·93 lbs. per sq. in. absolute, corresponding to 28 in. vacuum.		
	Velocity of outflow of steam, feet per second.	Kinetic energy, ft.-lbs. per sec.	H.P. of 550 ft.-lbs. per sec.	Velocity of outflow of steam, feet per second.	Kinetic energy, ft.-lbs. per sec.	H.P. of 550 ft.-lbs. per sec.	Velocity of outflow of steam, feet per second.	Kinetic energy, ft.-lbs. per sec.	H.P. of 550 ft.-lbs. per sec.
		Per lb. of steam per hour.			Per lb. of steam per hour.			Per lb. of steam per hour.	
60	2421	25·29	0·046	3320	47·57	0·087	3680	58·44	0·106
80	2595	29·06	0·053	3423	50·56	0·092	3793	62·08	0·113
100	2717	31·86	0·058	3520	53·47	0·097	3871	64·66	0·118
120	2822	34·37	0·062	3596	55·80	0·101	3940	66·99	0·122
140	2913	36·62	0·066	3661	57·84	0·105	3999	69·01	0·125
160	2992	38·63	0·070	3718	59·65	0·108	4045	70·61	0·128
180	3058	40·35	0·073	3764	61·14	0·111	4091	72·22	0·131
200	3115	41·87	0·076	3810	62·64	0·114	4127	73·50	0·134
220	3166	43·26	0·079	3852	64·03	0·116	4159	74·61	0·136
280	3294	46·83	0·085	3962	67·74	0·123	4229	77·18	0·140

Table IV., given by Mr. Konrad Andersson,[1] shows the

[1] Lecture delivered by Mr. Konrad Andersson at the Yorkshire College, January 13th, 1902.

velocity of outflow and rate of generation of kinetic energy of dry saturated steam expanding freely in suitable divergent nozzles under various conditions of pressure.

With the angle of the nozzle to the plane of the wheel fixed at 20 degrees, the best velocity for the vanes of the wheel neglecting friction is rather less than half the velocity of the steam jet. Practical considerations, however, usually cause the wheel to be run at a somewhat lower speed. Centrifugal force assumes enormous values, and it is only by careful design and construction, and the use of the best material, that De Laval turbine wheels can be run at the speeds they are.

Fig. 112 shows a 5-B.H.P. De Laval turbine wheel mounted on its shaft.

Figs. 113 and 114 are respectively front elevation and section of the same wheel drawn to a larger scale.

The stresses produced by centrifugal force on a rotating disc due to its mass were investigated in Chapter V. (pp. 73–75). In order that a greater velocity may be given

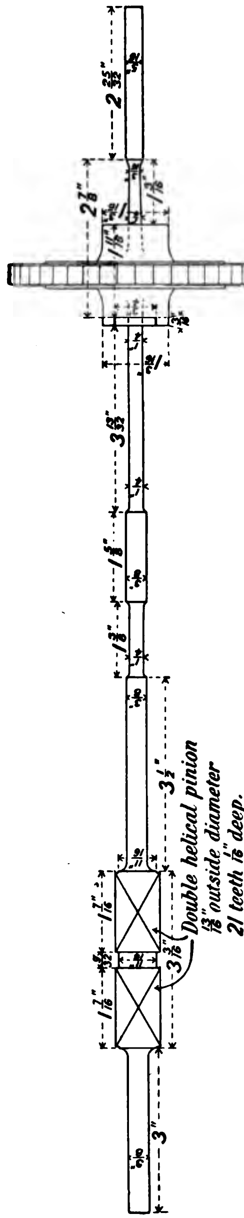


FIG. 112.—5-B.H.P. De Laval Turbine Wheel and Spindle.
(*The sizes are approximate only.*)

to a De Laval turbine wheel than could be given to a ring of the same diameter, the turbine wheel is constructed with a broad boss which tapers down towards the rim as shown. The material at the rim is supported not only by tangential forces, as in the case of a revolving ring, but also by radial forces; that is to say, the rim is to a certain extent hung from the

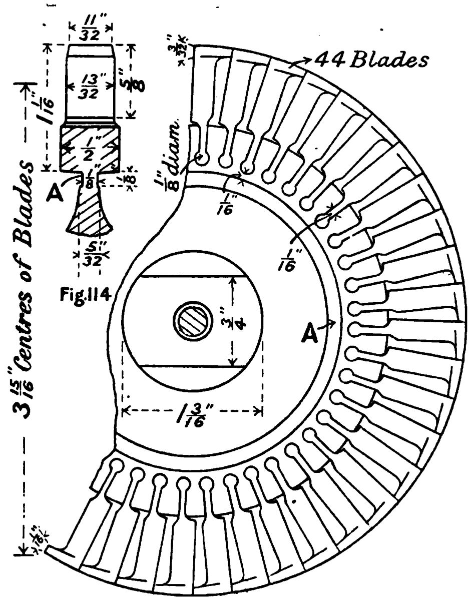


FIG. 113.—5-B.H.P. De Laval Turbine Wheel.
(The sizes are approximate only.)

boss. The centrifugal force per cubic inch of material at the boss is, of course, relatively small, and a considerable amount of material can therefore be placed there which will help to support the material at the rim where the centrifugal force is high. In the larger-sized turbine wheels the material cannot be spared from the centre of the boss, and in consequence the latter is not bored through, but the flexible shaft made in two portions and attached by means of hat-shaped flanges to the

two sides of the boss, as shown in Fig. 117. The wheel is purposely made weakest near the rim, as shown at A in Figs. 113 and 114. If the wheel breaks here due to stresses set up by centrifugal force, comparatively little damage will be done, while the loss of the blades will at once prevent any increase in the speed of the wheel.

Table V. gives the diameters and speeds of rotation of the turbine wheels of several sizes of De Laval turbines.

TABLE V.

DIAMETERS AND SPEEDS OF ROTATION OF SOME DE LAVAL TURBINE WHEELS.

Horse-power of turbine	5	30	100	300
Revolutions per minute	30,000	20,000	13,000	10,600
Diameter of wheel to centres of blades in inches	3·94	8·86	19·68	29·92

In the turbines of larger power the greater diameter of the turbine wheel allows of the same vane speed being attained with a less number of revolutions; but a greater vane speed is arranged for in the larger-power turbines. In some 300 H.P. De Laval turbines the normal mean speed of the vanes is about 1380 feet per second (15·6 miles per minute), and the normal speed of the extreme periphery of the wheel is, of course, slightly above this. As the blades are situated a mean distance of about 15 inches from the axis of rotation in these turbines, the centrifugal force on them amounts to nearly 21 tons per pound.* As each blade weighs about $\frac{1}{28}$ lb., the centrifugal force on each is about $\frac{3}{4}$ ton. The vane speed of the smaller De Laval turbines, such as 5 and 10 H.P., is usually only about 500 or 600 feet per second.

* $\text{C.F.} = \dfrac{mv^2}{r} = \dfrac{1 \times 1380^2}{32 \times \frac{15}{12}} = 47{,}600 \text{ lbs.} \doteq 20{\cdot}9 \text{ tons.}$

Fig. 115 is a longitudinal section of a blade made by a plane which is perpendicular to the axis of the turbine spindle. Fig. 116 is a cross-section of the blade. The method of dovetailing the blades into the rim of the wheel is shown in Fig. 115. The length of blade depends on the power of the

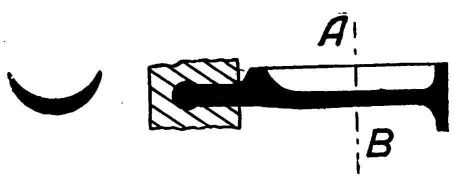


FIG. 116. FIG. 115.

Blade of De Laval Turbine.

turbine. A 5 H.P. turbine has blades $\frac{5}{8}$ inch long, while the blades of 300-H.P. turbine have been made $1\frac{1}{2}$ inches in length. A front view of a blade is shown in Fig. 114.

The pressure inside the turbine casing, F, Fig. 111, is practically that of the condenser or of the atmosphere, according as the turbine is condensing or non-condensing. It is desirable that this should be so, not only in order to increase the expansion of the steam in the nozzles, and thus deliver it on to the vanes at a higher speed, but in order to reduce the friction between the rotating turbine wheel and the fluid in which it rotates. This friction is found to be almost exactly proportional to the pressure inside the turbine-casing. If, through improving the condensing arrangement, the pressure inside the turbine casing can be halved, the work required to rotate the wheel against fluid friction would also be halved, if other conditions remained the same. The work required to overcome this friction when the turbine casing pressure is atmospheric is about fifteen times as much as when this pressure is only 1 lb. abs. under the same conditions as to speed and nature

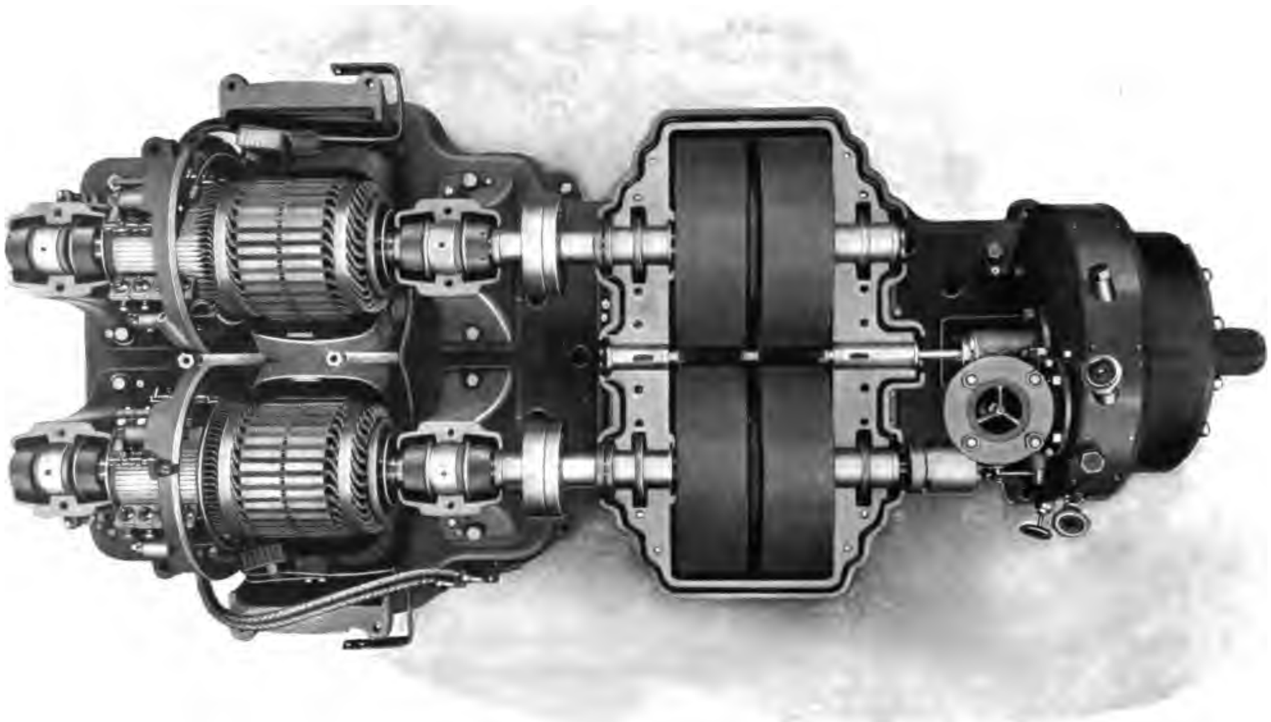


PLATE III.—110-H.P. DE LAVAL STEAM TURBO-DYNAMO—UPPER HALF OF GEAR CASE AND FIELD FRAME REMOVED.

of fluid. This fluid friction increases at a high rate with increase of velocity, and with the high speeds attained by the larger sizes of De Laval turbines it may attain huge dimensions. The advantages of condensing and of maintaining a good vacuum will therefore be evident.

The nature of the fluid also very much affects the friction. Superheated steam causes less friction than dry saturated steam, and this latter produces less friction than wet steam.

The expansion of the steam before it enters the vanes is also of great practical importance, for it allows of a considerable amount of clearance being permitted all round the turbine wheel in the case. In turbines constructed by the Société de Laval of France, a clearance of 2 to 5 millimetres is allowed all round the wheel. This permits of a very flexible shaft being used, as a slight displacement of the wheel may take place without any injurious consequences. This is very important, as the high speed at which the turbine wheel can run is dependent on the flexible nature of its support. A mass cannot rotate at the extreme speeds attained by the De Laval wheels without serious vibration unless it be allowed to rotate about its centre of mass. Now, it is found to be practically impossible to perfectly balance a wheel about any fixed axis. Even if the wheel be perfectly symmetrical in form about the axis, irregularities of density will upset the balance. The flexibility of the De Laval turbine spindle, however, allows the wheel to choose its own axis of rotation, and, after a certain critical speed has been passed, the axis of rotation chosen by the wheel is that which passes through its centre of mass. The wheel then runs perfectly smoothly. The critical speed is always a long way under the normal or designed speed of rotation of the turbine wheel.

The spindle can easily be made flexible, as its high rotary speed allows it to be of slender dimensions. A diameter of 1 inch is sufficient for the spindle of a 150-H.P. wheel, and a diameter of $1\frac{5}{16}$ inch suffices for that of a 300-H.P. wheel.

The flexible shaft is shown in Fig. 112, and also in Fig. 111. It has four journals, namely at Q, R, S, and T. The bearing for the journal S is self-aligning, and is held in place by a spring which can be seen in Fig. 111 (and also at H in Fig. 108). This bearing supports most of the weight of the turbine wheel. The bush which surrounds the journal T has perfect freedom to move with the shaft. The object of this bush is to prevent the flow of air into the turbine casing when the turbine is running condensing, or to prevent the flow of steam from the turbine casing when the turbine is running non-condensing. The bushes at Q and R take up the weight and thrust of the pinion J. The bush at Q (lettered F in Fig. 108) is always made in two parts. The bush at R (lettered D in Fig. 108) is sometimes made solid.

The pinion J, Fig. 111 (seen also in Fig. 112), is cut out of the metal of the turbine spindle (hard steel); but the wheel K (of somewhat softer steel) is forced upon the shaft L. The wheel and pinion are of the double helical type, and the teeth are small. Great accuracy is required in machining these teeth; but great strength is not necessary, as the forces exerted on the teeth are not excessive, owing to the high speed of rotation of the turbine shaft and the small diameter of the pinion. For example, the spindle of a 5-H.P. turbine makes 30,000 revolutions per minute, and the diameter of the pinion is about $\frac{3}{4}$ inch. The linear velocity of the teeth is therefore about 100 feet per second. If 6 horse-power is transmitted by the gearing, the tangential force exerted by the pinion on the wheel is only

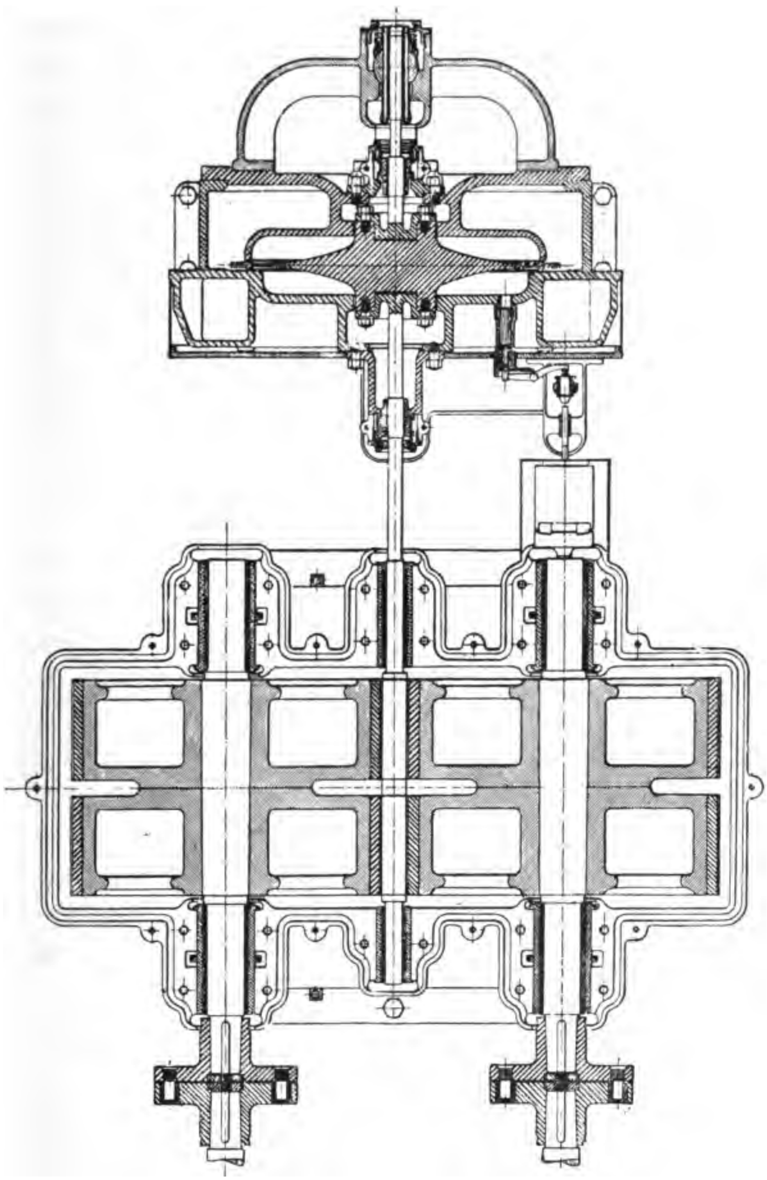


Fig. 117.—Sectional Plan of 300-H.P. De Laval Steam Turbine.

33 pounds, and, as this is always divided over a number of teeth, the pressure on each is obviously very small.

The larger De Laval turbines have two power shafts, each provided with a gear-wheel. The arrangement is clearly shown in Fig. 117 and also in Plate III., which shows a 110-H.P. turbo-dynamo constructed by the American De Laval Steam Turbine Company. It will be seen that the same pinion drives the gear-wheels of both power shafts, and that the power shafts rotate in the same direction.

For all sizes of De Laval turbines the gearing ratio is usually 10 to 1, and the linear velocity of the teeth about 100 feet per second. The teeth are cut by automatic machines. The gears work with great smoothness, and the wear on the teeth is found to be very slight.

The absence of reciprocating motion and the high speed of the gearing in a De Laval turbine obviate the necessity of extensive foundations such as are required by a reciprocating engine, and the smaller sizes of De Laval turbines require no foundations whatever.

The power shaft bearings are oiled by lubricating rings dipping into oil wells, as shown in Fig. 110. Sight feed lubricators are commonly employed for the bearings of the flexible shaft. The American De Laval Steam Turbine Company employ a central oil reservoir, U, Fig. 110, mounted upon the gear-case. This reservoir supplies lubricant to the bearings of both flexible shaft and power shaft.

As has already been stated, the speed of rotation of a De Laval turbine wheel has to be very great. Even the speed of the second motion or power shaft is usually very high. The great velocity and absence of pressure, however, allow of great lightness.

Table VI. gives the total weights of steam turbines of various sizes as made by the Société de Laval, with the angular velocities of the second motion or power shafts. The first five sizes have each one power shaft; the others have two.

TABLE VI.

WEIGHTS AND SPEEDS OF ROTATION OF DE LAVAL TURBINE MOTORS.

B.H.P. of turbine motor.	Total weight in kilograms.	Revolutions per min. of power shaft.
5	150	3000
10	225	2400
15	260	2400
20	420	2000
30	580	2000
50	1570	1500
75	1870	1500
100	2650	1250
150	3140	1040
200	4900	910
300	7650	775

A section of a De Laval governor as constructed by the Société de Laval (France) is shown in Fig. 118, and the parts

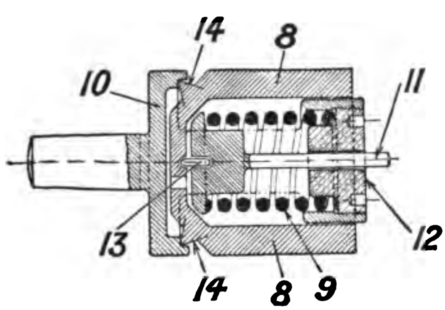


Fig. 118.—Section of Governor.

are shown separately in Fig. 119. The half cylinders 8, 8, are pivoted in the case 10 by the knife-edges 14, and have projecting

lugs which press on the spindle 11 through the agency of pins 13.

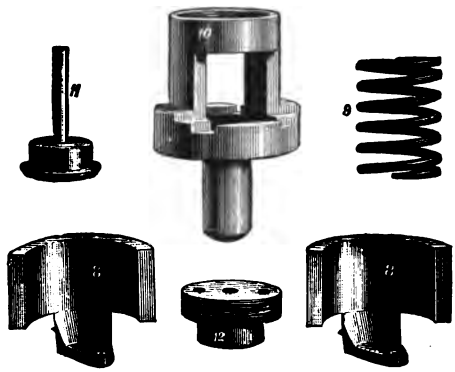


Fig. 119.—Parts of Governor.

Fig. 120 shows the half cylinders in their correct positions, but removed from the other parts. The spindle 11 acts by means of a lever on the steam admission valve. The centrifugal force is balanced by a spring, 9, which can be adjusted by means of the nut 12.

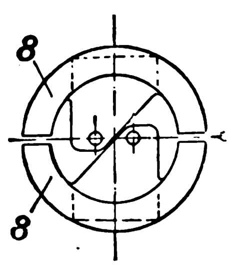


Fig. 120.—Half Cylinders of Governor in Position.

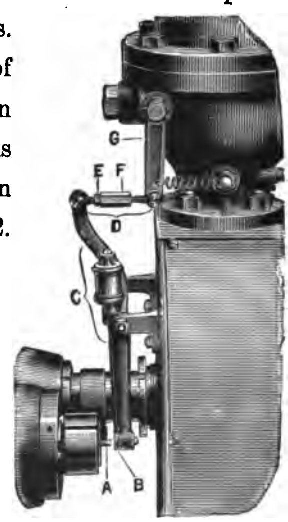


Fig. 121.—Connection of Governor with Steam Admission Valve.

The connection of the governor with the steam admission or throttle valve is shown in Fig. 121, where A is the spindle

which was marked 11 in Figs. 118 and 119. C is a lever pivoted near its centre, and arranged so that the spindle A can act on its lower end, while its upper end is connected to the lever G by means of a link which is adjustable by means of the nuts EF. The lever G operates the valve.

The governor used by the American De Laval Steam

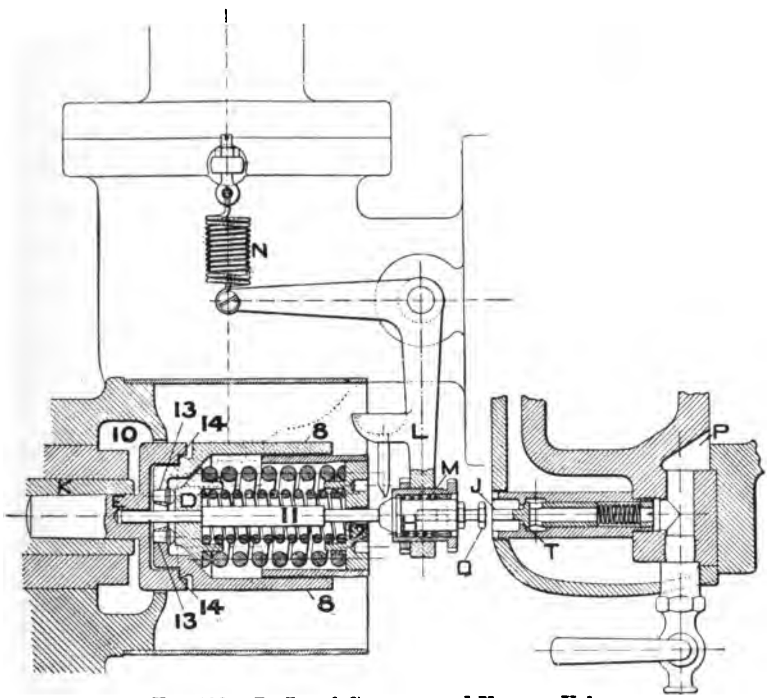


FIG. 122.—De Laval Governor and Vacuum Valve.

Turbine Company is shown in Fig. 122. It is of a similar nature to that just described, but two springs are employed instead of only one. The method of fitting the tapered projection E of the casing 10 into the end of the power shaft K is clearly shown. The spindle 11 in this case acts on a plunger, H, carried in the end of one arm of a bell-crank lever L. This

bell-crank lever can be moved against the action of the tension coil spring N so as to close the throttle valve, which is shown in Fig. 123. The governor, bell-crank lever and valve casing are shown in Fig. 110, where they are lettered O, P and C respectively.

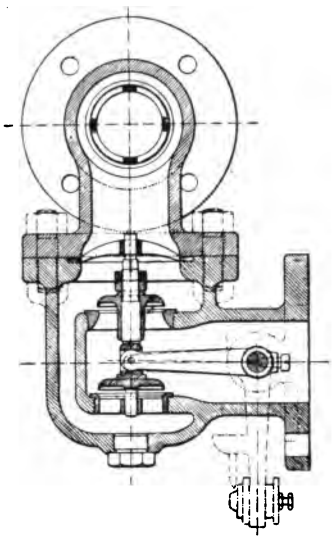


FIG. 123.—Governor Valve for De Laval Steam Turbine.

Fig. 122 also shows a vacuum valve which is used with condensing turbines in order to retain the speed within narrow limits. The plunger H is not rigidly fixed in the bell-crank lever, but is held in place by the spring M, which is stiffer than the spring N, so that the full movement is given to the bell-crank lever before any relative movement takes place between the plunger and the lever. When, however, the lever is pulled up by the throttle valve striking against its seat, the plunger H with any further movement of the spindle 11 strikes the end of the stem J of the vacuum valve T, opens this valve, and allows air from the atmosphere to flow into the turbine casing P. The connection of the vacuum valve to the turbine casing is also shown in Fig. 111. This admission of air not only diminishes the energy of the steam jets impinging on the blades of the wheel by reducing the ratio of expansion, but also acts as a powerful fluid brake to check the speed of the wheel.

The effectiveness of these governing devices will be appreciated by examining Figs. 124 and 125, which show speed

diagrams taken by means of a Horn's self-registering tachograph.

The governors above described are intended to act only for small or sudden changes of load as the reduction of steam pressure by throttling or the admission of air to the turbine casing of a condensing turbine, of course, adversely affect the efficiency. As far as possible, the changes of power of the

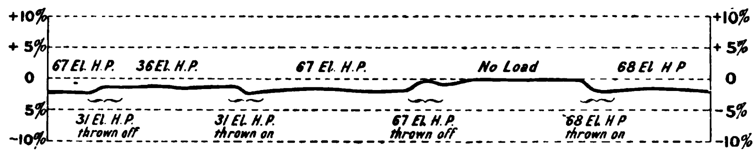


FIG. 124.—Speed variation diagram of De Laval Steam Turbine Dynamo, working non-condensing. The horizontal dotted line marked 00 represents a speed of 1050 revolutions per minute. The other dotted lines represent percentage increase or decrease of speed from this. The normal load is 135 El. H.P.

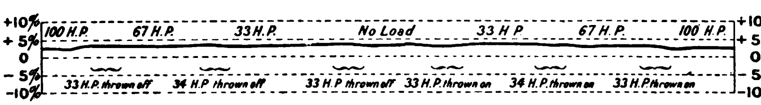


FIG. 125.—Speed variation diagram of De Laval Steam Turbine, working condensing. The horizontal dotted line marked 00 represents a speed of 1050 revolutions per minute. The other dotted lines represent percentage increase or decrease of speed from this. The normal load is 135 B.H.P.

turbine are affected by using a greater or less number of nozzles, the others being shut off by means of the hand wheels. A device is sometimes used whereby the nozzles are automatically opened or closed by means of steam actuated pistons.

When De Laval turbines are used for belt-driving, the pulleys placed on the turbine power shafts are of small diameter on account of the high speed at which these shafts rotate. A belt speed of about 5000 feet per minute is usually arranged for, as can be seen from Table VII., which gives the dimensions of pulleys employed by Messrs. Greenwood and Batley, Ltd. The smaller turbines have one power shaft and one pulley; the larger ones, two power shafts with a pulley on each.

TABLE VII.

Dimensions of Pulleys on De Laval Steam Turbines made by Messrs. Greenwood and Batley, Ltd.

B.H.P. of turbine.	Number of power shafts and pulleys.	Revolutions per minute of power shafts.	Diameter of pulleys in inches.	Width of face of pulleys in inches.
3	1	3000	$6\frac{1}{4}$	3
5	1	3000	$6\frac{1}{4}$	3
10	1	2400	8	4
15	1	2400	8	$4\frac{1}{2}$
20	1	2000	$9\frac{1}{2}$	5
30	1	2000	$9\frac{1}{2}$	6
50	2	1500	$13\frac{3}{4}$	7
75	2	1250	$15\frac{3}{4}$	$8\frac{3}{4}$
100	2	1050	18	$9\frac{3}{4}$
225	2	1000	$19\frac{15}{16}$	$17\frac{1}{8}$

The larger-sized turbines are sometimes arranged to drive by ropes, the diameters of the pulleys being about the same as for belt driving.

Table VIII. gives some particulars of steam turbine dynamos as constructed by the American De Laval Steam Turbine Company.

TABLE VIII.

Particulars of Turbine Dynamos as constructed by the American De Laval Steam Turbine Company.

B.H P.	K.W.	Revolutions per minute of armature spindles.	Length.		Width.		Approximate weight in lbs.	Diameter of steam pipe.	Diameter of exhaust pipe.
			ft.	in.	ft.	in.		in.	in.
$1\frac{1}{2}$	1	5000	2	6	0	11	250	$\frac{1}{2}$	1
3	2	3000	3	8	1	$4\frac{1}{4}$	470	$\frac{3}{4}$	$1\frac{1}{2}$
5	3·3	3000	4	1	1	10	850	$1\frac{1}{4}$	2
7	4·6	3000	4	2	1	10	900	$1\frac{1}{4}$	2
10	6·6	2400	5	0	2	1	1,550	$1\frac{1}{2}$	3
15	10	2400	5	3	2	2	1,720	$1\frac{1}{2}$	3
20	13·2	2000	6	3	2	7	2,100	2	4
30	20	2000	6	4	2	10	2,800	2	4
55	35	1500	8	1	3	3	5,000	$2\frac{1}{2}$	5
75	50	1500	8	7	3	10	9,000	$3\frac{1}{2}$	6
110	75	1200	9	9	4	7	13,000	4	8
150	100	1200	10	10	4	8	16,000	4	8
225	150	900	12	11	5	11	23,000	5	10
300	200	900	15	0	6	3	30,000	5	12

Machines from 50 to 200 K.W. are fitted with double generators.

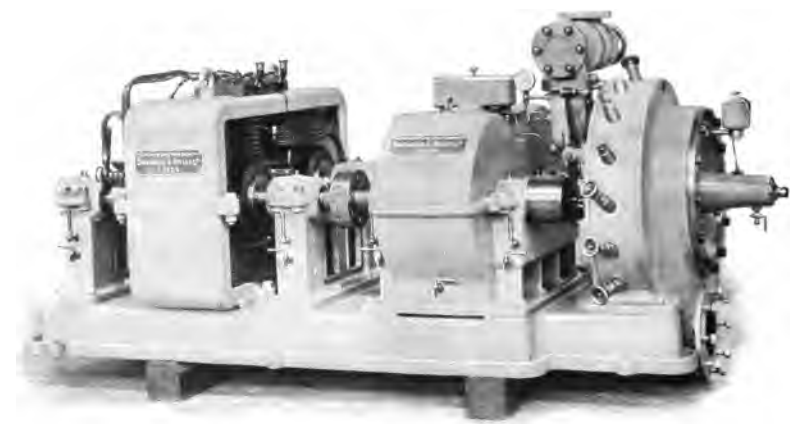


Plate IV.—100-B.H.P. De Laval Turbine Dynamo made by Messrs. Greenwood and Batley, Leeds.

Plate IV. shows 100 B.H.P. De Laval steam turbine dynamo made by Messrs. Greenwood and Batley of Leeds. This machine has been delivered to, and is now at the works of, the Morris Aiming Tube and Ammunition Co., Ltd., Essex. It will be seen that there are two armatures. These are mounted on shafts which carry inside the gear-box helical-toothed wheels, which gear one on each side with a pinion mounted on the turbine shaft. The turbine wheel case is seen at the right of the figure, with the wheels for controlling the supply of steam to the several nozzles. The flange for connection to the steam supply is seen over the turbine case, and the exhaust outlet is shown at the bottom of the case. The weight of this machine complete is 6 tons, and the designed speed of rotation of the armature-spindles is 1050 revolutions per minute. A machine of the same power as this, but of earlier design, has been in constant use for several years at the Albion Works of Messrs. Greenwood and Batley.

Plate V shows a 300 H.P. De Laval steam turbine and dynamo constructed by the American De Laval Steam Turbine Company. The turbine casing, gear-case, dynamo field frame and shaft pedestals are mounted on a single bed-plate which is about 14 feet long and about 6 feet in greatest width. One field frame is used for the two dynamos. This is divided horizontally to allow of the convenient removal of the armatures. As the field frames used with these turbines are of somewhat unusual appearance, one is shown in Fig. 126. It will be seen to resemble a figure 8. The pole pieces, of which it will be seen there are eight in all, are secured to the field frame each by two bolts. The yokes for carrying the brush-holders are also well shown in Fig. 126. These yokes are cast-

iron rings supported on brackets cast integral with the field frame. The yokes can each be rotated through a certain angle by means of a screwed rod and hand wheel in order to adjust the angular position of the brushes. The latter are borne by

Fig. 126.—Double Field Frame and Brush Holder Yokes for De Laval Turbine Dynamo.

current-carrying brackets fixed to but insulated from the yokes. The division of the field frame and the gear-case horizontally can be well seen in Plate III.

Figs. 127–134, and Tables IX.–XII. give the chief dimensions of steam turbine dynamos from 3 to 300 H.P., as made by the American De Laval Steam Turbine Company.

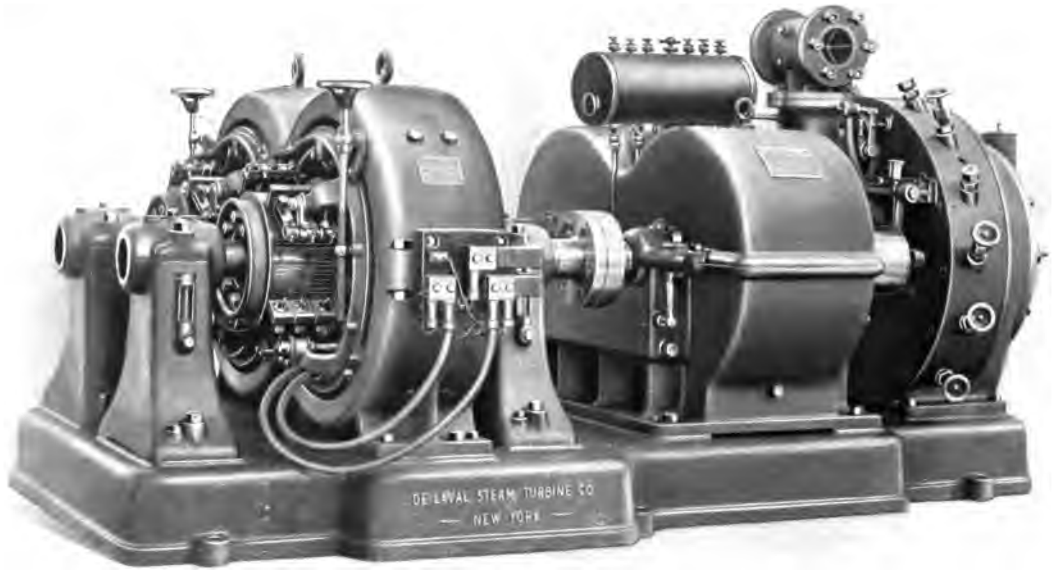


PLATE V.—300-H.P. DE LAVAL STEAM TURBINE AND DOUBLE DYNAMO.

DIMENSIONS OF DE LAVAL STEAM TURBINES FROM 3 TO 7 H.P.

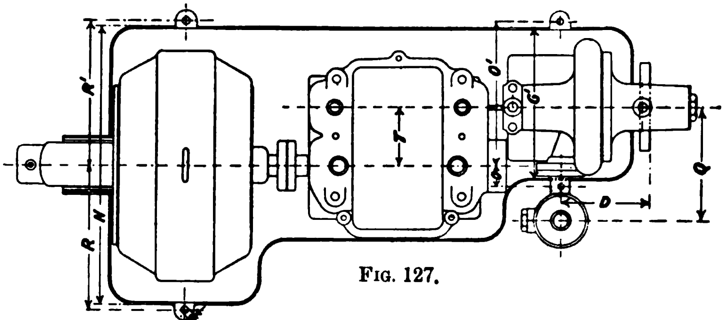


FIG. 127.

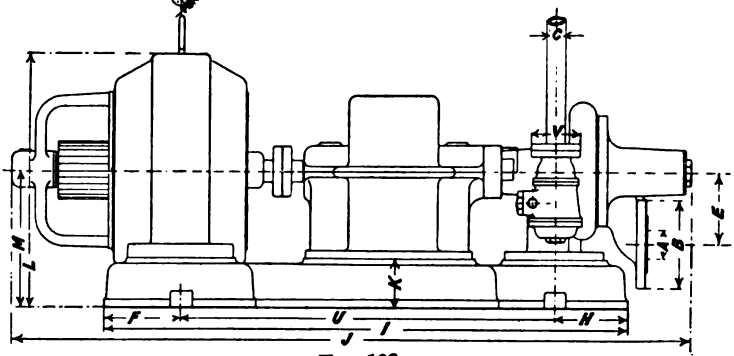


FIG. 128.

TABLE IX.

Dimensions in inches.

	3 H.P.	5 H.P.	7 H.P.		3 H.P.	5 H.P.	7 H.P.
A	$1\frac{1}{2}$	2	2	M	$9\frac{1}{4}$	$9\frac{1}{4}$	$9\frac{1}{4}$
B	6	6	6	N	$15\frac{1}{2}$	$19\frac{3}{8}$	$19\frac{3}{8}$
C	$\frac{3}{4}$	$1\frac{1}{4}$	$1\frac{1}{4}$	O	$1\frac{15}{16}$	$1\frac{15}{16}$	$1\frac{15}{16}$
D	$6\frac{5}{16}$	$6\frac{5}{16}$	$6\frac{5}{16}$	O¹	$8\frac{3}{16}$	$8\frac{3}{16}$	$8\frac{3}{16}$
E	$4\frac{15}{16}$	$4\frac{15}{16}$	$4\frac{15}{16}$	Q	$8\frac{1}{8}$	$8\frac{1}{8}$	$8\frac{1}{8}$
F	$5\frac{1}{16}$	$7\frac{1}{8}$	$7\frac{1}{8}$	R	$8\frac{3}{16}$	$10\frac{1}{8}$	$10\frac{1}{8}$
G	$9\frac{1}{4}$	$9\frac{1}{4}$	$9\frac{1}{4}$	R¹	$8\frac{3}{16}$	$10\frac{1}{8}$	$10\frac{1}{8}$
H	$4\frac{1}{4}$	$4\frac{1}{4}$	$4\frac{1}{4}$	S	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{5}{8}$
I	$35\frac{5}{16}$	$39\frac{11}{16}$	$40\frac{15}{16}$	T	4·3295	4·3295	4·3295
J	$44\frac{3}{4}$	$48\frac{5}{8}$	$49\frac{7}{8}$	U	26	$28\frac{5}{16}$	$29\frac{9}{16}$
K	$3\frac{1}{4}$	$3\frac{1}{4}$	$3\frac{1}{4}$	V	$3\frac{3}{4}$	$4\frac{3}{4}$	$4\frac{3}{4}$
L	$15\frac{3}{4}$	$17\frac{3}{8}$	$17\frac{3}{8}$				

DIMENSIONS OF DE LAVAL STEAM TURBINES FROM 10 TO 20 H.P.

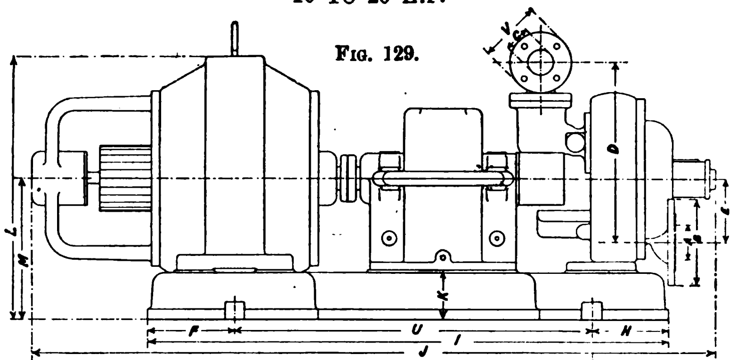


FIG. 129.

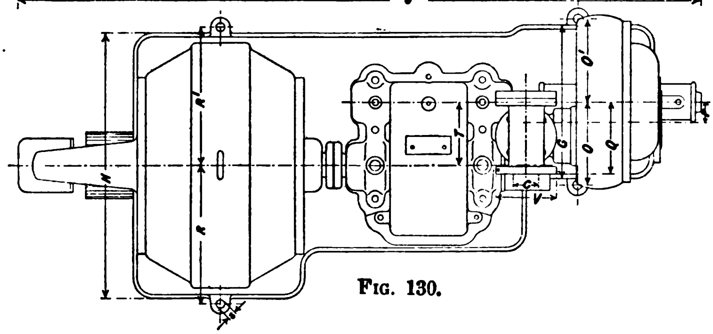


FIG. 130.

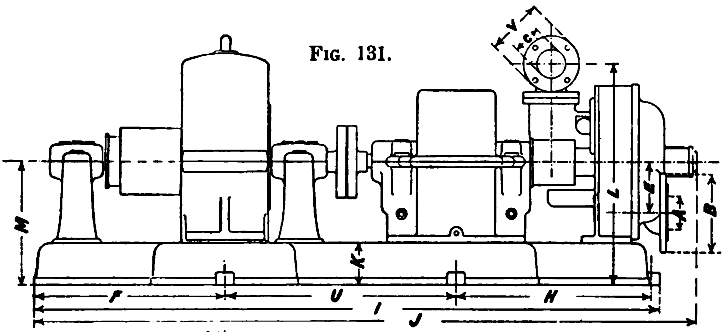
TABLE X.

Dimensions in inches except where otherwise stated.

	10 H.P.	15 H.P.	20 H.P.		10 H.P.	15 H.P.	20 H.P.
A	3	3	4	M	$11\frac{5}{8}$	$12\frac{1}{8}$	14
B	$7\frac{1}{2}$	$7\frac{1}{2}$	9	N	$22\frac{1}{8}$	$25\frac{1}{8}$	$25\frac{1}{4}$
C	$1\frac{1}{2}$	$1\frac{1}{2}$	2	O	$7\frac{3}{8}$	$7\frac{7}{16}$	$10\frac{1}{4}$
D	$15\frac{3}{8}$	$15\frac{3}{8}$	17	O¹	$4\frac{5}{16}$	$4\frac{3}{8}$	$5\frac{3}{8}$
E	$5\frac{1}{4}$	$5\frac{1}{4}$	$5\frac{3}{4}$	P	$1\frac{1}{2}$	$1\frac{1}{2}$	$2\frac{3}{4}$
F	$7\frac{1}{4}$	$11\frac{1}{4}$	$8\frac{1}{2}$	Q	$6\frac{1}{8}$	$6\frac{1}{8}$	$6\frac{15}{16}$
G	$10\frac{11}{16}$	$10\frac{13}{16}$	$14\frac{5}{8}$	R	$11\frac{9}{16}$	$13\frac{1}{16}$	$13\frac{1}{8}$
H	$4\frac{9}{16}$	$4\frac{11}{16}$	$4\frac{7}{8}$	R¹	$11\frac{9}{16}$	$13\frac{1}{16}$	$13\frac{1}{8}$
I	47	$52\frac{7}{16}$	53	S	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$
J	$4'\ 11\frac{1}{2}''$	$5'\ 2\frac{1}{2}''$	$6'\ 2\frac{3}{8}''$	T	5·527	5·527	7·75
K	$3\frac{3}{4}$	$4\frac{1}{4}$	$4\frac{3}{4}$	U	$35\frac{3}{16}$	$36\frac{1}{2}$	$39\frac{5}{8}$
L	$20\frac{5}{8}$	$22\frac{3}{4}$	$24\frac{5}{8}$	V	$5\frac{5}{16}$	$5\frac{5}{16}$	6

DIMENSIONS OF DE LAVAL STEAM TURBINES OF 30 AND 55 H.P.

FIG. 131.



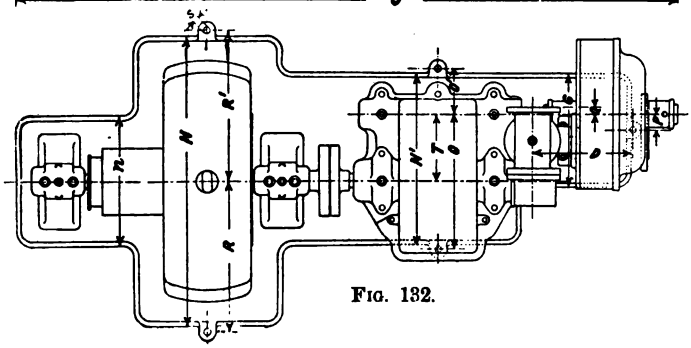


FIG. 132.

TABLE XI.

Dimensions in inches except where otherwise stated.

	30 H.P.	55 H.P.		30 H.P.	55 H.P.
A	4	5	N	32	36
B	9	$10\frac{1}{4}$	N¹	$19\frac{7}{8}$	$23\frac{11}{16}$
C	2	$2\frac{1}{2}$	n	$15\frac{3}{4}$	18
D	$10\frac{3}{4}$	$10\frac{5}{8}$	O	$16\frac{1}{8}$	$19\frac{3}{4}$
E	$5\frac{3}{4}$	$6\frac{1}{8}$	O¹	$4\frac{3}{4}$	$5\frac{1}{16}$
F	$22\frac{7}{16}$	$26\frac{3}{4}$	P	$2\frac{3}{4}$	$2\frac{11}{16}$
G	14	$14\frac{3}{8}$	Q	$1\frac{1}{16}$	$1\frac{1}{16}$
H	$22\frac{1}{8}$	$26\frac{1}{2}$	R	$16\frac{1}{2}$	$18\frac{9}{16}$
I	$71\frac{1}{2}$	$92\frac{5}{8}$	R	$16\frac{1}{2}$	$18\frac{9}{16}$
J	6′ 4″	8′ $1\frac{3}{8}$″	S	$\frac{3}{4}$	$\frac{7}{8}$
K	5	7	T	7·757	10·195
L	$25\frac{7}{16}$	29	U	$26\frac{1}{32}$	$38\frac{3}{8}$
M	$14\frac{1}{4}$	$17\frac{3}{4}$	V	6	7

TABLE XII.

DIMENSIONS OF DE LAVAL STEAM TURBINES FROM 75 TO 300 H.P.

Dimensions in inches except where otherwise stated.

	75 H.P.	110 H.P.	150 H.P.	225 H.P.	300 H.P.
A	6	8	8	10	12
B	11	$13\frac{1}{2}$	$13\frac{1}{2}$	16	18
C	$3\frac{1}{2}$	4	4	5	5
D	$22\frac{3}{8}$	$25\frac{7}{16}$	$25\frac{7}{16}$	40	$40\frac{11}{16}$
E	$14\frac{3}{4}$	20	20	$24\frac{1}{4}$	$28\frac{1}{4}$
F	$5\frac{1}{4}$	$6\frac{13}{16}$	$7\frac{3}{8}$	$8\frac{5}{16}$	$9\frac{11}{16}$
G	24	$28\frac{1}{4}$	$28\frac{1}{4}$	$42\frac{5}{8}$	$48\frac{7}{8}$
H	$7\frac{1}{2}$	$8\frac{11}{16}$	$8\frac{11}{16}$	$10\frac{23}{32}$	$11\frac{11}{16}$
I	$8'\ 0\frac{3}{4}''$	$9'\ 2\frac{3}{8}''$	$10'\ 3''$	$11'\ 9\frac{9}{16}''$	$13'\ 10\frac{7}{8}''$
J	$8'\ 6''$	$9'\ 8\frac{3}{4}''$	$10'\ 9\frac{5}{16}''$	$12'\ 10\frac{7}{8}''$	$15'\ 0''$
K	7	8	8	9	10
L	$35\frac{1}{8}$	$39\frac{1}{4}$	$39\frac{1}{4}$	$45\frac{3}{4}$	48
M	20	23	23	$26\frac{1}{2}$	$28\frac{3}{4}$
N	$46\frac{1}{16}$	$51\frac{3}{4}$	$55\frac{3}{4}$	$5'\ 10\frac{5}{8}''$	$6'\ 3''$
O	$12\frac{3}{4}$	15	$15\frac{1}{8}$	$22\frac{5}{16}$	$25\frac{15}{16}$
O^1	$12\frac{3}{4}$	15	$15\frac{1}{8}$	$22\frac{5}{16}$	$25\frac{15}{16}$
Q	$8\frac{5}{16}$	$9\frac{3}{8}$	$9\frac{3}{8}$	$14\frac{1}{8}$	$15\frac{1}{4}$
R	$18\frac{25}{32}$	$22\frac{7}{8}$	$24\frac{7}{8}$	$29\frac{9}{16}$	$32\frac{1}{16}$
R^1	$18\frac{25}{32}$	$22\frac{7}{8}$	$24\frac{7}{8}$	$29\frac{9}{16}$	$32\frac{1}{16}$
r	$16\frac{13}{16}$	$18\frac{1}{2}$	$21\frac{1}{2}$	$23\frac{1}{16}$	$27\frac{3}{16}$
r^1	$16\frac{13}{16}$	$18\frac{1}{2}$	$21\frac{1}{2}$	$23\frac{1}{16}$	$27\frac{3}{16}$
S	1	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{5}{8}$
U	$45\frac{5}{16}$	$44\frac{25}{32}$	53	$60\frac{17}{32}$	$67\frac{25}{32}$
U^1	$38\frac{11}{16}$	$50\frac{1}{8}$	$53\frac{7}{8}$	$62\frac{1}{16}$	$77\frac{3}{4}$
V	$8\frac{1}{2}$	$8\frac{1}{2}$	$8\frac{1}{2}$	$10\frac{3}{4}$	$10\frac{3}{4}$

De Laval steam turbines have been successfully employed in the lighting of trains on the Prussian State Railways. A De Laval non-condensing turbine is mounted on the same bed-plate as an enclosed dynamo, and is secured on top of the boiler of the locomotive. Mains lead from the dynamo brushes along the length of the train, and are connected to a battery of accumulators in each car. Each glow lamp is provided with a resistance which limits the voltage between the lamp

terminals to 48 volts. In some instances a 20 B.H.P turbine is used, and in this case the dynamo makes 2000 revolutions per minute. The steam consumption of the turbine has been stated to be in some cases from 42 to 44 lbs. per B.H.P. hour. (Non-condensing.)

DE LAVAL STEAM TURBINES FROM 75 TO 300 H.P.

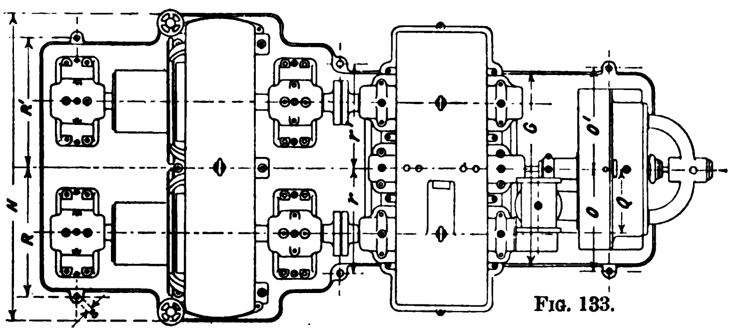


FIG. 133.

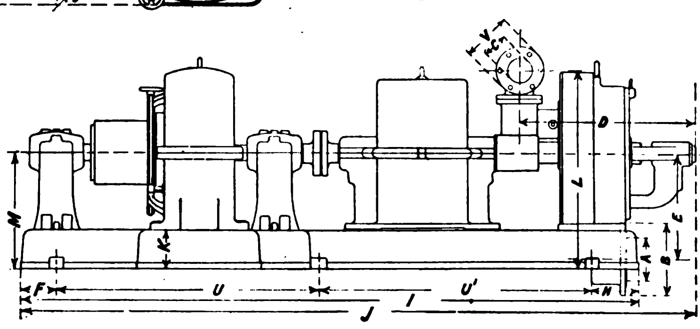


FIG. 134.

Plate VI. shows a 200 kilowatt two-phase turbo-alternator constructed by the American De Laval Steam Turbine Company.

Plate VII. shows a 300 H.P. De Laval Turbine Dynamo, constructed by the American De Laval Turbine Company, and of somewhat different design to that shown in Plate V.

Plate VIII. shows a 225 B.H.P. De Laval turbine constructed by Messrs. Greenwood & Batley, Ltd., and provided with pulleys for belt driving. One pulley, $19\frac{5}{16}$ inches in diameter and $17\frac{1}{8}$ inches wide, is arranged on each power shaft which rotates at 1000 revolutions per minute. This gives a belt speed of about 5000 feet per minute. The pulleys are arranged clear of each other lengthwise so that both can drive in the same direction.

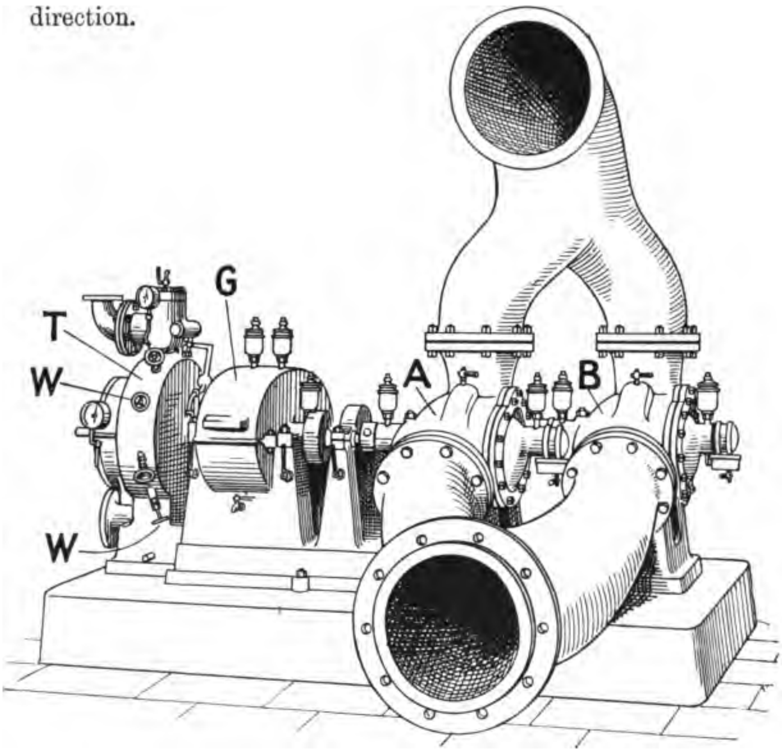


FIG. 135.—De Laval Turbine (Parallel) Centrifugal Pump.

Fig. 135 shows a De Laval turbine centrifugal pump as supplied by the same firm. T is the turbine-wheel casing, surrounding which can be seen the wheels W for controlling the steam jets. G is the gear-wheel casing, emerging from

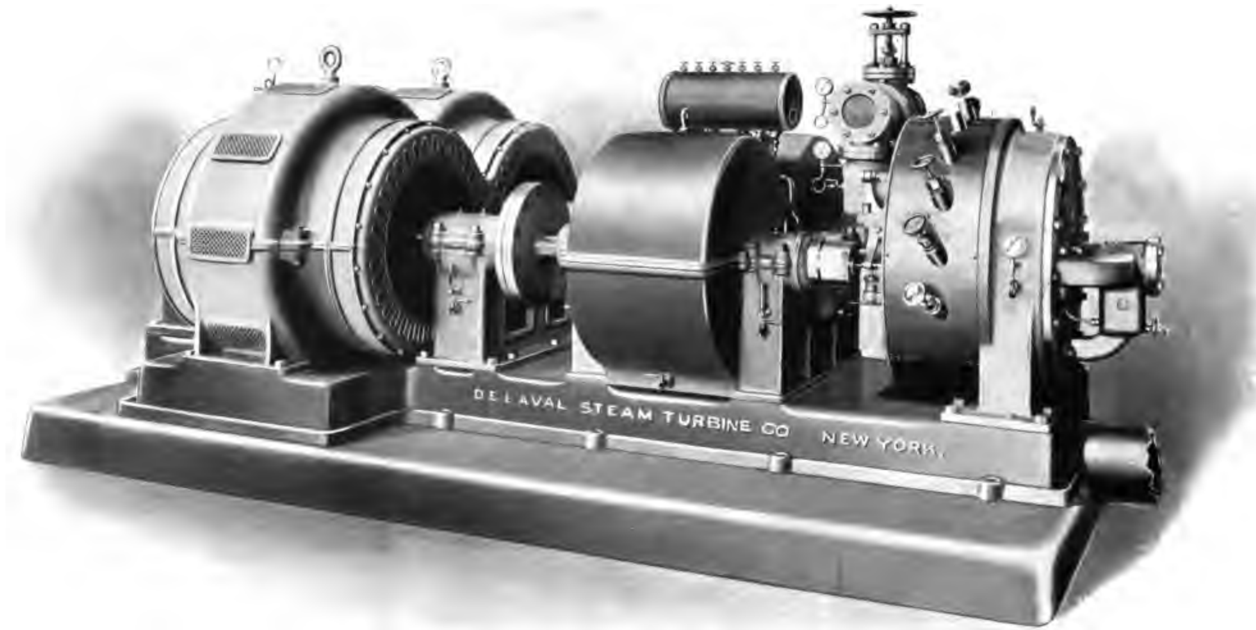


Plate VI.—200-K.W. Two-Phase De Laval Turbo-Alternator.

which can be seen the two pump shafts, which are driven like the armature shafts of a turbine dynamo. Centrifugal pumps A and B are arranged one on each shaft, the two pumps working in parallel.

Fig. 136 shows a turbine pump constructed by the Société de Laval, but in which the pumps are arranged in series as regards delivery of water. The pumps are arranged on parallel shafts as in the pump just mentioned, but in the series pump the delivery end of the one pump is connected to the suction end of the other, so that the pressure and not the amount of water is doubled.

The constant turning moment on the pump shaft causes these pumps to work very quietly, and the high speed of rotation allows the water to be raised to considerable heights.

When it is desired to deliver water to great heights or against great pressures, an arrangement is sometimes adopted in which two pumps are employed—one driven direct from the turbine spindle and the other through gearing. The water is first dealt with by the slower speed pump and then passed on to the direct driven one. A compound pump of this nature has delivered 640 gallons of water per minute against a head of 467½ feet. Water can be delivered against considerably greater heads by employing a number of pumps in series.

Fig. 137 shows a turbine blower as constructed by the Société de Laval.

Table XIII. gives particulars of tests made by Messrs. Erik Andersson, Karl Wallin, and Axel Estelle, at the works of the Aktiebolaget de Laval's Angturbin in Sweden in 1895, on a 50 H.P. turbine dynamo. Steam was generated at 118 lbs. per square inch, and reduced by a throttle valve. The turbine had 6 induction nozzles.

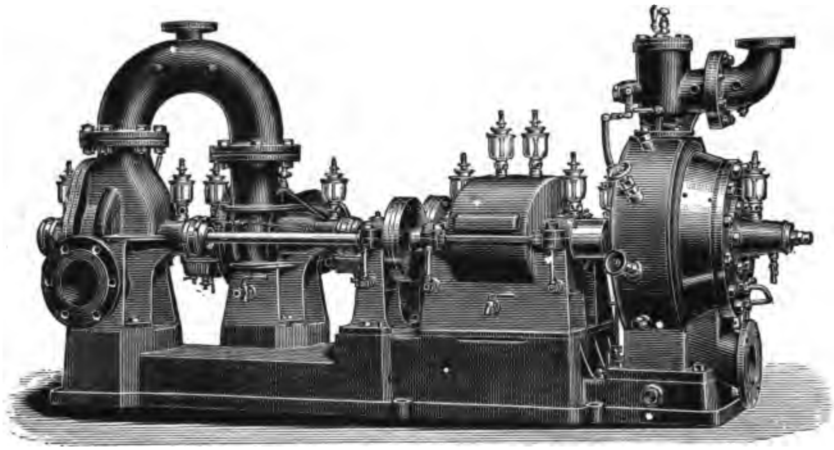


FIG. 136.—De Laval Turbine (Series) Centrifugal Pump.

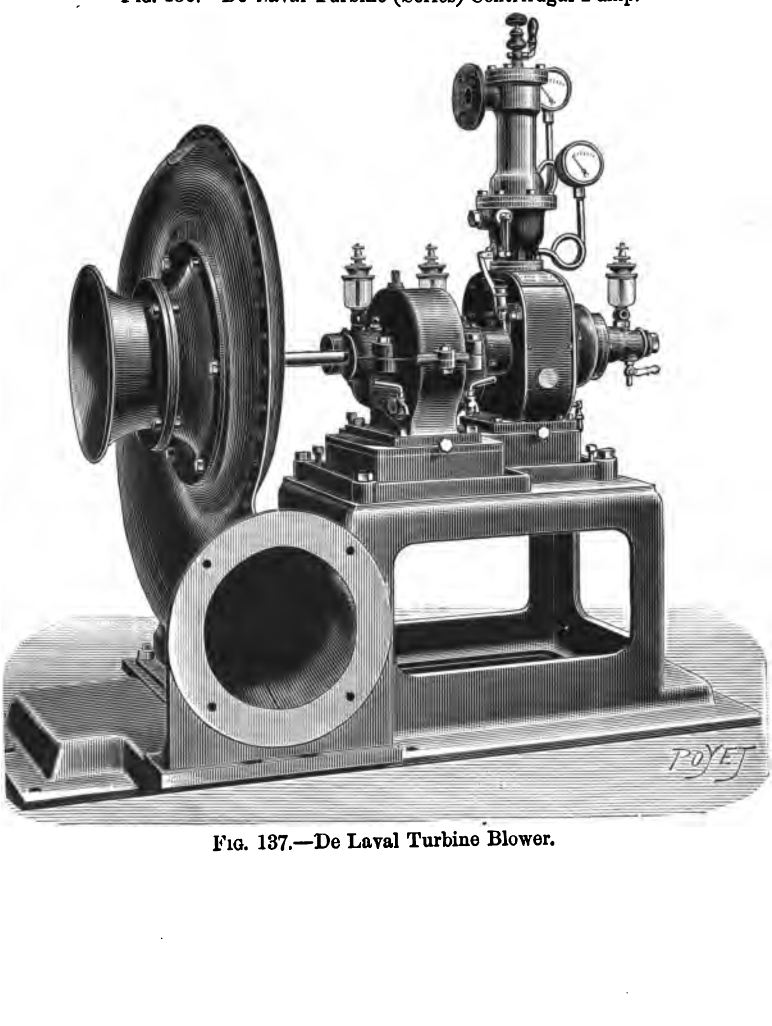


FIG. 137.—De Laval Turbine Blower.

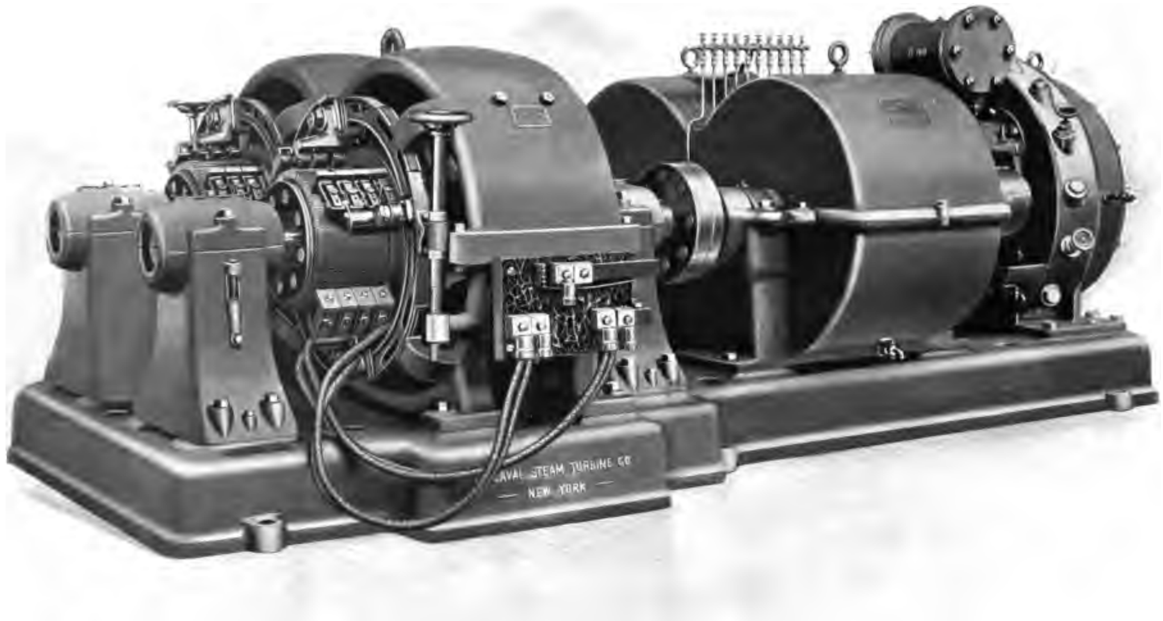


PLATE VII.--300-H.P. DE LAVAL STEAM TURBINE AND DOUBLE DYNAMO.

TABLE XIII.

TEST OF DE LAVAL TURBINE DYNAMO.

Date of trial.	E.H.P. $\frac{\text{volts} \times \text{amp.}}{736}$	Steam pressure; lbs. per square inch.	Vacuum; lbs. per square inch.	Number of nozzles used.	Lbs. of steam per E.H.P. per hour.
Feb. 15	49·92	114	13	6	24·5
Mar. 4	50·05	114	—	6	24·2
"	40·79	114	—	5	24·76
"	21·72	114	—	3	27·9
"	25·34	93·8	13·27	4	27·49
"	12·87	74	13·5	3	32·0

It should be noted that the electrical horse-power unit is obtained by dividing the product of volts and ampères by 736 instead of by 746, as is done in this country. The steam consumption was obtained on March 4, by inserting one of the steam nozzles into a pipe leading to a vessel containing a quantity of water where the steam was condensed. The amount of steam passing through this nozzle was thus ascertained, and it was assumed that the amount passing through each of the other (acting) nozzles was the same, the design and cross-sections of the nozzles being identical. The amounts probably were nearly the same, as was shown by a check test, but it cannot, of course, be assumed that this will be so in every case.

Table XIV. shows the results of tests on a De Laval turbine made by Professor Cederbolm, of Stockholm, in November, 1897. The power was measured by a brake.

TABLE XIV.

DE LAVAL TURBINE OF 150 BRAKE HORSE-POWER.

No. of nozzles used.	Brake horse-power.	Steam pressure.		Vacuum.		Revolutions.	Consumption of steam per B.H.P. per hour.	
		Kilos per sq. centim.	Lbs. per square inch.	Millim. of mercury.	Inches of mercury.		Kilos.	Lbs.
7	165·3	8·00	113	670	26·4	1057	7·87	17·3
5	116·1	8·00	113	666	26·2	1057	8·01	17·6
3	65·0	7·90	112	685	27 0	1060	8·49	18·7

In 1896 tests were made of the steam consumption of one of the turbine dynamos supplied by the Société de Laval to the Edison Electric Illuminating Company of New York. The tests were made at the works of the Illuminating Company in New York.

The following is a summary of the trial, the report on which is signed by Messrs. Smith, Van Vleck, and De Kermel representing the Edison Electric Illuminating Company, and Mr. Paré, who represented the Société de Laval.

Duration of trial	6 hours.
Mean steam pressure	10 kilos. per square centimetre, or 143 lbs. per square inch.
Mean vacuum in condenser	65 centimetres, or $25\frac{1}{2}$ inches.

Dynamo No 1, mean volts 127·25, mean ampères 708·56.
Dynamo No. 2, mean volts 128·26, mean ampères 727·47.

The total power generated was therefore about 183 kilowatts.

A surface condenser was used, and was tested to prove that no leakage took place. The amount of water condensed in the six hours was 12,493·35 kilograms, or 2082·22 kilograms per hour.

The steam consumption per kilowatt hour was therefore $\frac{2082\cdot22}{183}$, or 11·38 kilograms (*i.e.* 25·1 lbs.).

Seven hundred and thirty-six watts were taken as an electrical horse-power, and the efficiency of the dynamo was assumed to be 90 per cent. The brake horse-power of the turbine under these assumptions was therefore about 276·7, and the consumption of steam per B.H.P. per hour $\frac{2082\cdot22}{276\cdot7}$, or 7·52 kilograms (*i.e.* 16·6 lbs.).

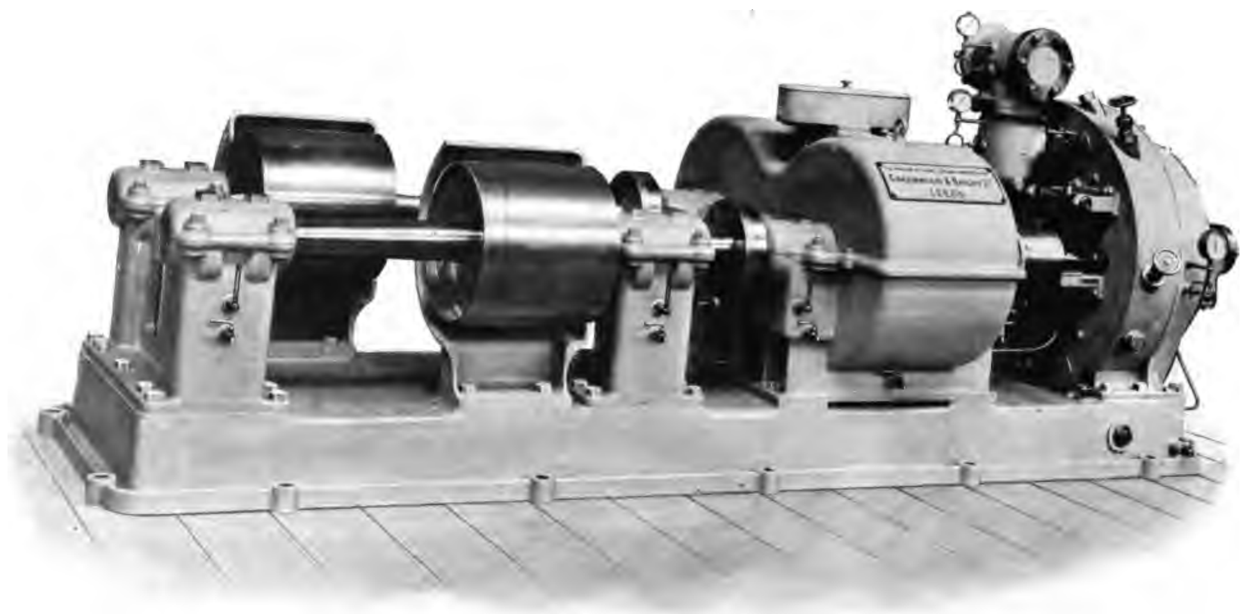


Plate VIII.—225-H.P. De Laval Steam Turbine constructed by Messrs. Greenwood and Batley, Ltd.

Tests were made in May and June, 1902, of a De Laval Steam Turbine at the works of the De Laval Steam Turbine Company at Trenton, N.J., U.S.A. This turbine is employed to drive two dynamos which supply direct current for actuating the tools in the machine shop; but during some of the tests a water rheostat was employed to absorb some of the electrical energy, as the tools did not require all the current generated.

The feed water supplied to the boiler was measured by weighing in a barrel in order to ascertain the steam consumption. When superheated steam was used, a bare stem Green thermometer was inserted in a well of cylinder oil in the steam admission pipe. When saturated steam was used, the amount of moisture it contained just as it entered the turbine casing was determined by a Peabody throttling calorimeter and an allowance made for this. An allowance was also made for water withdrawn from the steam by a separator on the line of steam pipe; but no allowance was made for condensation which occurred in the chamber supplying the turbine nozzles (D in Fig. 111).

The tests were conducted by Messrs. Dean and Main of Boston, U.S.A. The efficiencies of the generator at different loads were ascertained. The electrical instruments were checked at the works of the Weston Electrical Instrument Company, Waverly Park, N.J., and the brake horse power computations were made by Messrs. Stone & Webster of Boston, and by the De Laval Electrical Engineer.

Particular note should be taken of the result of the last test —16·40 lbs. of dry saturated steam per B.H.P. hour *at about one-third load.*

TABLE XV.

SUMMARY OF TESTS OF 300 B.H.P. DE LAVAL STEAM TURBINE AT TRENTON, N.J., U.S.A.

Duration of Test in Hours.	Average Steam Pressure before Governor Valve in lbs. per sq. in.	Average Steam Pressure after Governor Valve in lbs. per sq. in.	Average Vacuum in inches of Mercury.	Barometer in inches of Mercury.	Average Temperature of Room in degrees Fahr.	Averg. Superheat before Governor Valve in deg. Fahr.	Number of Nozzles open.	Revolutions per minute of Generators.	Average Brake Horse-Power.	Lbs. of Steam per Hour.	Lbs. of Steam per B.H.P. Hour.
6	207·0	198·5	27·2	30·18	83	81	8	750	352	4906	13.94
2	207·4	197·0	27·4	30·07	90	64	7	756	298·4	4282	14.35
6	201·5	197·2	27·4	29·79	89	13	5	745	196·2	3047	15·53
4	206·4	196·9	26·6	29·92	90	0	8	747	333·0	5052	15·17
2	207·3	196·5	26·8	29·90	97	0	7	746	284·8	4430	15·56
2	207·6	·195·8	27·3	29·83	97	0	5	751	195·2	3229	16·54
3	201·5	197·9	28·1	29·81	80	0	3	751	118·9	1950	16·40

Superheating the steam very much improves the efficiency of a De Laval steam turbine; it allows a brake horse power to be obtained for a fewer number of heat units supplied by the steam. The improvement is in all probability chiefly due to the reduction of friction within the turbine casing. Steam superheated to as high as 500° C. (932° F.), has been tried with good results on a 30 H.P. De Laval turbine working non-condensing, but sufficient tests do not seem to have been made to determine whether the gain with this great superheat is worth the cost of obtaining it. In the experiment in question the temperature of the exhaust steam was 343° C. (649° F.)

De Laval steam turbines are manufactured under the De Laval Patents in England by Messrs. Greenwood and Batley, Ltd., of Leeds; in the United States of America by the De Laval Steam Turbine Company of New York; in France by

the Société De Laval of Paris; in Sweden by the Aktiebolaget de Lavals Angturbin of Stockholm, and in Germany by the Maschinenbau-Anstalt Humboldt of Kalk bei Koln.

CHAPTER IX.

THE RATEAU STEAM TURBINE.

It has already been pointed out that in the Parsons turbine the steam is expanded gradually in passing alternately through fixed and moving rings of blades, while in the De Laval type of turbine, the steam is expanded in a divergent nozzle before it reaches the vanes of a single rotating wheel. It has also been pointed out that the De Laval type of motor has an advantage over the Parsons type in so far that the amount of clearance round the wheel does not need to be small. In a Parsons turbine, the clearance spaces beyond the free ends of the fixed and moving blades require to be small to prevent excessive leakage of steam, especially at the high-pressure end of the turbine. It is true that the leakage of steam round the rings of blades, instead of through between the blades, does not represent the same loss of power as the leakage of steam past the piston of a reciprocating engine, for the steam that leaks past one ring of blades reserves its energy for the next ring, or gives up its saved heat to the rest of the steam. It will, however, be obvious on a little consideration that this leakage of steam must entail a lengthening of the turbine cylinder, and an increase in the number of rings of blades, in order to expand the steam to the desired extent. This involves increase in bulk, weight, cost, and radiation.

To minimize the leakage of steam, great accuracy and good workmanship are required; and, although these requisites can be commanded by Messrs. C. A. Parsons and Co., they might not be obtained in less well-equipped or less well-managed works.

The disadvantage of the De Laval type of steam turbine is the excessive velocity which the blades must have, necessitating the use of gearing to obtain speeds of rotation which can be utilized for industrial purposes. In the more powerful De Laval motors, the larger turbine wheels employed allow somewhat smaller angular velocities to be obtained without reducing the velocity of the vanes; but in all cases the number of revolutions per minute of the turbine wheel is very high. Apart from the objectionable feature of gearing, the velocity of the vanes is limited by the strength of the materials of construction. As has already been pointed out in Chapter V., the stress due to centrifugal force in a rotating ring becomes enormous at high velocities. The vane-speed in a De Laval turbine is thus limited by the strength of material obtainable. Increasing the diameter of the wheel makes approximately no difference if the vane-speed remains constant. By arranging the vanes on the periphery of a disc which increases in thickness from the circumference to the centre, a somewhat higher speed may be obtained, the inner parts of the disc supporting the outer, but still a limit is reached to the safe speed before the best velocity is obtained for utilizing the energy of high-pressure steam.

The Rateau steam turbine, as now constructed by Messrs. Sautter, Harlé and Co., of Paris, and by the Maschinenfabrik Oerlikon of Switzerland, has been devised with the object of obtaining the advantages of the De Laval motor while adopting

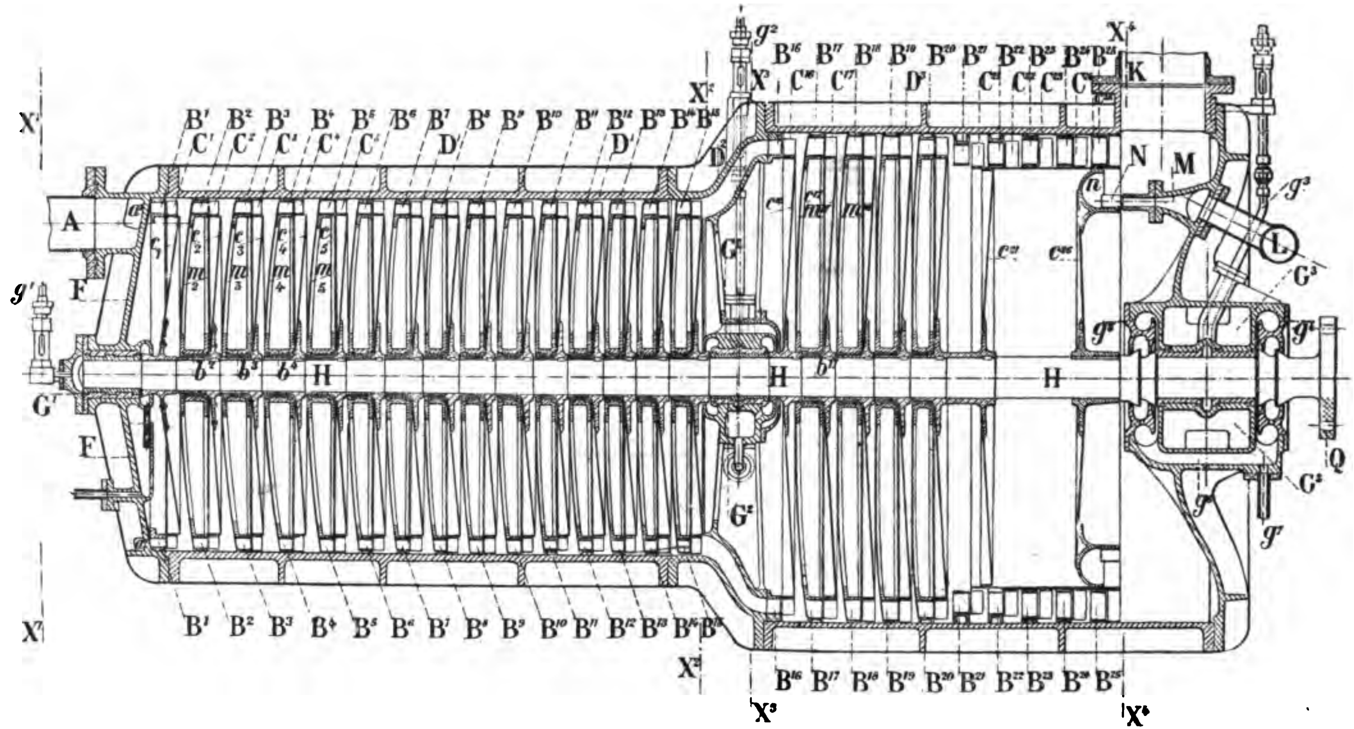
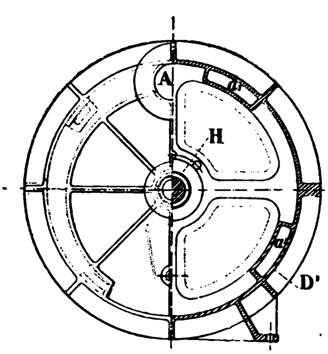


FIG. 138.—Rateau Steam Turbine: Longitudinal section.

the plan of expansion in steps and action on a series of wheels in order to obtain a more moderate speed of rotation.

This type of steam turbine is somewhat like the Parsons parallel flow-motor, but differs from the latter in this respect, that each rotating ring of blades revolves, as it were, in a compartment by itself. If we can imagine a number of De Laval, turbines placed side by side with the wheels in parallel planes, and if we imagine a large number of nozzles extending from the exhaust side of one wheel, through the casings to the steam side of the next wheel, we have in principle a Rateau turbine as now constructed at Oerlikon and by Messrs. Sautter Harlé and Co., of Paris.

Fig. 138 is a longitudinal section through a marine steam turbine of this type, and Figs. 139 and 140 are transverse



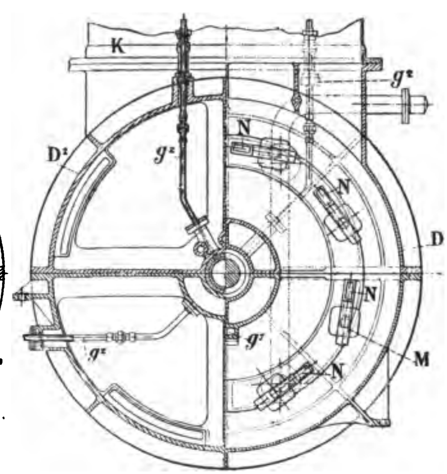


FIG. 139. FIG. 140.
Transverse Sections of Rateau Steam Turbine.

sections of the same. The plane of section of the left-hand half of Fig. 139 is represented by the line X^1X^1 on Fig. 138, and the plane of section of the right-hand half by the line X^2X^2.

The planes of section of the left and right-hand parts of Fig. 140 are represented on Fig. 138 by the lines X^3X^3 and X^4X^4 respectively.

The cast-iron or cast-steel cylinder D^1, D^2, D^3 is made in several parts and is strengthened by circumferential ribs. The high-pressure end of the cylinder is closed by the dished plate F to a flange on which is attached the steam-pipe A. Steam passages, a^1, a^1, are provided to allow of the steam reaching the first distributor or guide ring B^1. This distributor consists of a series of blades which occupy a portion of the inner circumference of the casing. These blades guide the steam in the proper direction on to the blades C^1 of the first rotating disc c^1. This disc is of thin steel slightly dished and is attached to an annular flange formed on a hub mounted on the turbine shaft H. The disc is formed with a circumferential flange to which the blades are attached. Fig. 141 shows a method of riveting the rotating blades C to the flanged periphery P of the disc, two consecutive blades being shown. The pieces U are cast on to the blades at their flanged ends to stiffen them. It will be seen from the figures that the arrangement is very light. The steam on passing the rotating blades C^1 enters a chamber enclosed between the disc c^1 and a diaphragm m^2. This diaphragm extends from a hub, b^2, which surrounds the shaft without touching it to the distributing blades B^2. These blades are fixed to a casting which is attached steam-tight to the inside of the cylinder. The construction is such that the steam can enter the chamber only by way of the rotating blades C^1, and can leave the chamber only by way of the distributing blades B^2. These distributing blades direct the steam on to the second set of rotating blades C^2, after passing

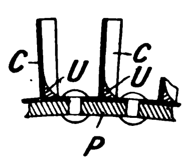


FIG. 141.

through which the steam enters another chamber enclosed between the disc c^2 and diaphragm m^3, which is attached to the distributing blades B^3 and to the hub b^3, the diaphragm, distributing blades, and hub being similar to the preceding, except that the area allowed for the passage of steam is greater. The construction is continued in a similar manner to the end of the cylinder. The diameter of the cylinder is increased at D^2 to afford greater area to the steam. Any steam that may leak out between any set of distributing blades and the succeeding rotating blades is in a closed chamber between the diaphragm of this set of distributing blades on the left hand and the diaphragm of the next set of distributing blades on the right hand. Each revolving disc, therefore, rotates in a closed chamber just like a De Laval turbine, the pressure on both sides of the rotating disc being approximately the same. There is, of course, a slight leakage of steam between the hubs b^2, b^3, etc., and the shaft; but a clearance of a few millimetres here does not allow a large area for the escape of steam, and any distortion of the machine is not so likely to cause rubbing at this point as at the circumference of the rotating discs.

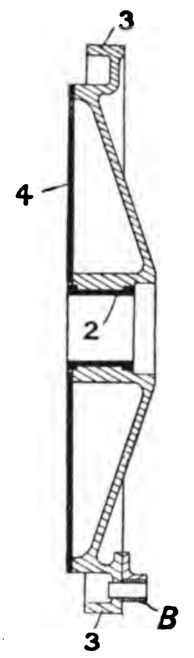


Fig. 142. — Diaphragm and Distributing Vane of Rateau Turbine.

Fig. 142 shows one of the diaphragms attached to the distributing blades of a slightly different design to that shown in Fig. 138. The bush 2 inserted in the hub is just clear of the shaft. The part 3 fits into a groove in the surrounding cylinder. One of the distributing vanes is

shown at B, these vanes being usually fitted only on a small part of the circumference at the high-pressure end of the turbine and increasing in number towards the low-pressure end; 4 is a plate riveted on the front of the diaphragm in order to present a smooth surface to the steam and so reduce friction.

The last five rings of rotating blades C^{21} to C^{25} are not mounted like the others, but are attached to the exterior of a drum which is connected to the shaft H by the discs c^{21} and c^{25}. The distributing blades B^{21} to B^{25} are connected only to the enclosing cylinder. K is the exhaust passage, and it will therefore be seen that the back of the plate c^{25} is exposed to the pressure of the exhaust, while the front of the plate c^{21} is exposed to the pressure of the steam which acts on the blades C^{21}. An axial thrust is thus exerted on the rotating parts of the turbine, and this axial thrust is used to wholly or partially balance the thrust of the screw propeller. In a turbine used for driving a dynamo or otherwise where no axial thrust is required, this arrangement of blades which is the same as that in a Parsons parallel-flow turbine may be dispensed with. The arrangement has the disadvantage, mentioned at the beginning of this chapter, that leakage of steam will take place between the inner periphery of the distributing rings of blades and the exterior of the drum carrying the rotating blades.

For rotating the turbine in a reverse direction (when this is required) a number of vanes, N, are provided, the curvature of which is opposite to that of the other moving vanes. Steam is guided on to these vanes by nozzles, M, leading from a supply conduit L. The rotating vanes, N, are carried by the disc c^{25}, and the steam exhausting from these vanes is guided by the annular trough *n* to the exhaust passage K.

The shaft H is supported in three bearings G^1, G^2, G^3. These bearings are supplied with oil under pressure conveyed to the respective bearings by the pipes g^1, g^2, g^3. The pressure of oil in the bearing G^3 is used to prevent air leaking into the exhaust end of the turbine when the latter is connected to a condenser.

M. Rateau has also experimented, in conjunction with Messrs. Sautter, Harlé and Co., with other types of steam turbines having one, two, or more discs. Some account of these experiments is given in M. Rateau's interesting paper presented to the International Congress on Applied Mechanics held at Paris in the summer of 1900. In a one-disc turbine described in the paper, the disc was formed from a single piece of special forged steel in the periphery of which the vanes were milled. These vanes were of the double type like those of a Pelton water-wheel. The disc increased in thickness from the periphery to the centre, this design being adopted for the sake of strength to resist centrifugal force. The stress produced in a ring due to the centrifugal force caused by its own weight, has already been discussed in Chapter V. M. Rateau has calculated that, by substituting a disc of uniform thickness for a ring, the allowable speed of rotation is only increased by 7 per cent. In this case the dangerous part is at the centre of the disc. By increasing the thickness progressively towards the centre, however, M. Rateau was able to run his wheels at considerably higher speeds; and with special hard steel he has obtained, without rupture, peripheral velocities of 400 metres per second.

The steam was projected on to the disc by several nozzles, and the jets of steam were divided in two by the central ridges of the buckets, as in the Pelton water-wheel. The nozzles

were arranged to project the steam on to the lower part of the disc so that the impact of the steam helped to balance the weight of the disc. One or more of the nozzles could be put out of action to decrease the power of the motor. The best results were obtained by supporting the disc shaft in one bearing only. The disc had thus a slight play owing to the flexibility of the shaft, and was able to choose its own centre of rotation. Gearing was employed to reduce the speed of the disc, the gear-wheels being of double helicoidal form and enclosed in a dust-proof box. The best form of packing tried at the place where the shaft passed through the side of the casing containing the disc consisted of a ring split into three pieces along three diametrical planes. This split disc was pressed against the side of the casing by means of springs. When the shaft did not vibrate, the ring worked as if solid, whilst, when the shaft did vibrate, the three pieces moved apart to give it freedom. This packing was found to be tight as long as the vibrations of the shaft were not considerable.

M. Rateau has recently patented a governing device for multiple-expansion steam turbines. According to this device, a small increase or decrease of speed from the normal (or desired) speed of the turbine causes a centrifugal governor to act on the steam admission valve. A greater *increase* of speed causes a relay cylinder to act on an obturator which admits steam to a greater or less number of the passages through the first guide ring in the turbine casing. The governor, however, still continues to act on the steam admission valve. A greater *decrease* of speed acts to open a valve admitting high-pressure steam to an intermediate part of the turbine casing, the governor in addition still acting on the steam admission valve.

CHAPTER X.

FURTHER REMARKS ON THE PARSONS TURBINE.

THE efficiency of a condensing steam turbine depends largely on the pressure at the exhaust end, or, in other words, the number of inches of vacuum at this end. This fact was referred to in Chap. IV., p. 59. Tables XVI. and XVII. show the effect on a Parsons turbine of altering the vacuum in the condenser. It will be seen that every additional inch of vacuum reduces the steam consumption about 4 per cent.

TABLE XVI.

CONSUMPTION OF 500-KILOWATT PARSONS TURBO-ALTERNATOR RUNNING AT 2500 REVOLUTIONS WITH 140 LBS. STEAM PRESSURE AT THE STOP-VALVE AND NO SUPERHEAT. (Based on results of tests.)

Vacuum constant from full load to no load.	Consumption of steam per kilowatt-hour.			Consumption per hour.
Inches of mercury.	full load.	half load.	quarter load.	no load.
29	—	—	—	1500
28	22·2	25·6	32·4	1700
27	23·1	26·9	34·5	1900
26	24·0	28·2	36·6	2100
25	25·1	29·7	39·0	2300
24	26·2	31·2	41·2	2500
23	27·5	32·9	44·8	2700
22	28·9	34·7	46·3	2900

Barometer = 30 ins.

TABLE XVII.

CONSUMPTION OF 1000-KILOWATT TURBO-ALTERNATOR CONSTRUCTED BY MESSRS. C. A. PARSONS AND CO. FOR ELBERFELD CORPORATION. NO SUPERHEAT.

Pressure stop-valve.	Vacuum. Barometer = 30 ins.	Kilowatts.	Steam per kilowatt-hour.
lbs. per sq. in.	inches of mercury.		lbs.
157·5	26·97	1010	23·08
153·0	24.45	1041	25·25
125·0	27·10	1022	20·47

But even with a good vacuum in the condenser, the efficiency may be spoilt by the throttling of the exhaust by narrow pipes or passages between the turbine and the condenser. To prevent any possibility of this, Mr. Parsons has invented

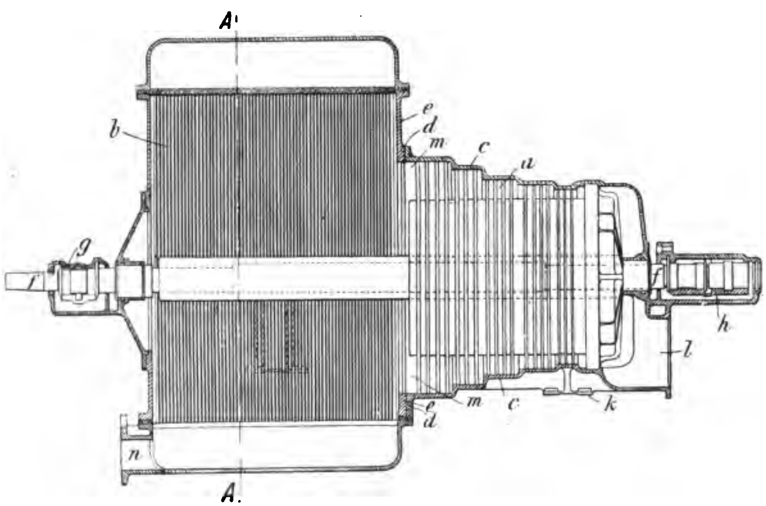


FIG. 143.—Vertical section.
Parsons Combined Turbine and Condenser.

and patented a combined turbine and condenser, which is illustrated in Figs. 143, 144, and 145, the turbine being of the parallel-flow type. Fig. 143 shows the combination in

vertical section, and Fig. 144 in plan; while Fig. 145 is a partial vertical section on the line AA of Fig. 143. The casing *c* of the turbine is bolted at *d* to the end *e* of the condenser.

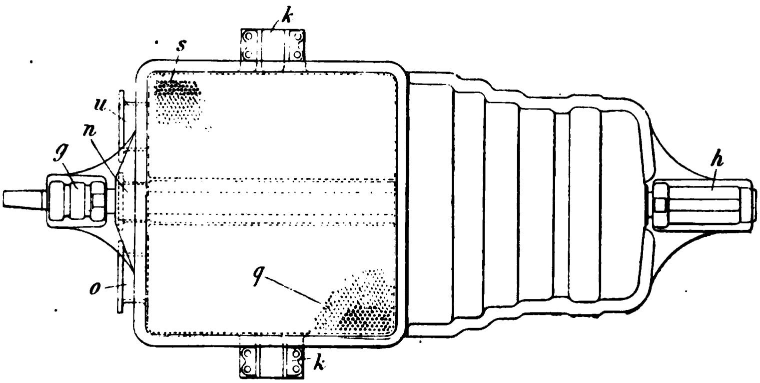


FIG. 144.—Plan.
Parsons Combined Turbine and Condenser.

The turbine spindle *f* passes through the turbine and condenser, and is supported in bearings at *g* and *h*. The turbine and condenser casings are supported on feet, *k*, *k*. The steam enters the turbine casing at *l*, and, after going through the fixed and moving rings of blades *a*, passes directly out of the large end of the turbine casing into the condenser. The turbine and condenser casings act as if made in one piece, and are, in fact, only made in separate castings for constructional reasons. The outlet from the condenser to the air-pump is shown at *n*. The circulating water enters the condenser at *o*, and passes into compartment *p*. It then passes up through the tubes *q* to the

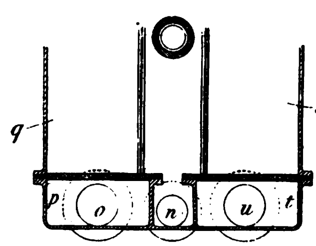


FIG. 145.—Partial vertical section on line AA of Fig. 143.
Parsons Combined Turbine and Condenser.

top chamber r, whence it descends through the tubes s to the compartment t, and leaves the condenser at u.

If the condenser is separate from the turbine as is the usual practice, the diameter of the connecting pipe for the exhaust steam has to be considerable if full advantage is to be derived from the great expansion obtainable in a Parsons turbine. The large size of the exhaust-pipe will be noticed in Fig. 60 and in subsequent illustrations, and the necessity for this will be seen by referring back to Fig. 70.

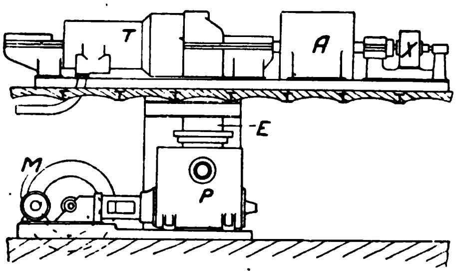


FIG. 146.

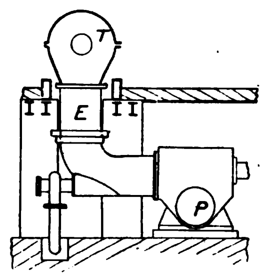


FIG. 147.

Arrangement of Steam Turbine, Alternator and Condenser.

It is also desirable that the condenser should be as near as possible to the exhaust end of the turbine, and it is often placed directly below this, and in a basement below the floor on which the turbine rests. Figs. 146 and 147 illustrate such an arrangement as used with a Parsons turbine constructed by Messrs. Brown, Boveri and Co. The exhaust-pipe E leading from the low-pressure end of the turbine T passes through the concrete floor to the condenser. The air-pump P and the electro-motor M which drives it rest on the floor of the basement. A is the alternator (1500 K.W.) driven by the turbine, and X is the exciter.

When it is desired to cause the shaft of a turbine to revolve

in a reversed direction, this is usually accomplished by placing a reversing turbine on the same shaft as the main turbine. The main and reversing turbines are usually in separate casings, and steam is admitted to one or the other according to the direction of rotation desired. Both have their exhaust ends permanently connected to the condenser, so that the one not working rotates in the condenser vacuum; and, as there are no rubbing parts within the casing of a turbine, the drag of the inoperative turbine is almost inappreciable.

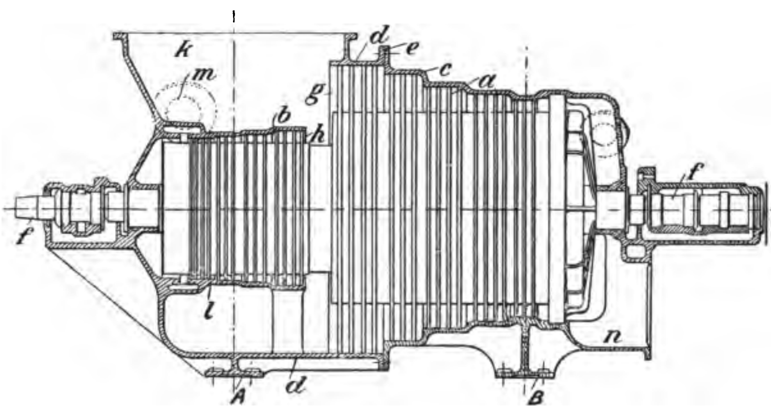


FIG. 148.—Parsons Arrangement of Main and Reversing Turbines in One Casing.

The main and reversing turbines may, however, with advantage be placed within the same casing. Fig. 148 shows an example of this. The main turbine *a* is enclosed chiefly in the casing *c*, which is bolted at *e* to the casing *d*. The casings *c* and *d* are cast with feet, A and B, and the casing *d* also carries an internal cylinder, *l*, which encloses the reversing turbine *b*. Both turbines are of the parallel-flow type, and both have their moving rings of blades attached to the spindle *f*. The low-pressure ends *g* and *h* of the two turbines open into the passage *k*, leading to or forming a part of the condenser. The

steam supply for the main turbine enters by the passage *n*, while that for the reversing turbine is admitted through the casing *d* at *m*. The turbines shown in Fig. 148 are intended for marine purposes, and the reversing turbine is therefore smaller than the main turbine, as the astern speed of a vessel is not usually required to be so great as the ahead speed.

Fig. 149 shows another example of main and reversing turbines in one casing. The reversing turbine *b* is here telescoped within the main turbine to save longitudinal space.

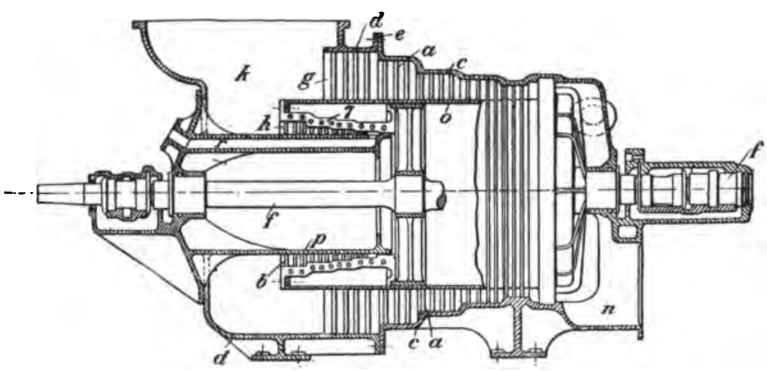


Fig. 149.—Parsons Arrangement of Telescoping Reversing Turbine within Main Turbine.

The stationary rings of vanes of the main turbine are fixed, as is usual, to the casing *c*, the moving rings being attached to the drum *o*, which is fixed to the shaft *f*. The reversing turbine has its fixed rings of blades attached to the exterior of the cylinder *p*, which is fixed to the casing *d*, while its moving rings are carried by the casing 7, which is rigid with the drum *o*. The steam enters the main turbine at *n*, while it gains access to the reversing turbine by the pipe *r*. The exhaust ends, *g* and *h*, of both turbines open directly into the condenser passage *k*.

A Parsons turbine can be reversed by interchanging its steam and exhaust connections so that the steam passes through the turbine in the reverse direction, but the efficiency is not as great. If the blades are designed for maximum efficiency in one direction, the efficiency when rotating in the other direction is much reduced. The usual construction of blades has been already shown in Figs. 3, 4, 5, 8, 10, and 69. It will be seen that both in the fixed and in the moving blades the space between two adjacent blades converges from the side at which the steam enters to the side at which it leaves. The concave face of the blade at the side at which the steam enters is almost

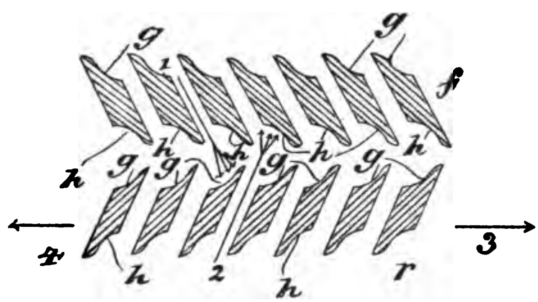


FIG. 150.—Form of Blades adapted for Rotating in Either Direction.

at right angles to the direction of motion of the moving blades. When the flow of steam is reversed, the blades are much less efficient. Mr. Parsons has patented the form of blade illustrated in Fig. 150 for use in turbines intended to run in both directions. Here the blades are straight for the greater part, but each blade is hollowed out at both ends, at *g* and *h*, so that, whichever way the steam flows, it impinges on a concavity. The fixed blades are lettered *f*, and the revolving blades *r*. The latter move in the direction of the arrow 3 when the steam passes as indicated by the arrow 1, and in

the direction of the arrow 4 when the steam flows as indicated by the arrow 2.

Messrs. C. A. Parsons and Co. have a very ingenious machine for constructing the rings of blades used in their turbines. The rings of blades are usually constructed with a heavy shroud at one end and a light shroud at the other, the heavy shroud being inserted and caulked into a groove in the turbine casing or revolving drum as shown in Fig. 6, p. 4. Shrouds of suitable metal, preferably brass, are formed into a circle, or segment of a circle. On one edge of the strip, teeth of special shape are cut by means of a circular cutter. The form of the teeth is such that, when the blades are laid in the grooves and the teeth turned over them, the teeth and blades fit each other closely, and form a secure fastening. This will be clear by referring back to Figs. 3, 4, and 5. In Fig. 3 some of the teeth of the shrouds are shown before they are bent down over the blades. The bending down of the teeth is performed by a punch which acts about three or four teeth behind the cutting-tool, so as to give the attendant time to insert the blades. The blades are, however, commonly attached to the light shroud only by a copper wire.

The governing of a Parsons turbine is usually effected by varying the duration of puffs or blasts of steam admitted to the turbine. Fig. 151 shows an electrical governor arranged for this purpose. The solenoid U is energized by electric current (it is a shunt to the field magnets of the electric generator driven by the turbine), so that increase or decrease of speed of the turbine causes the lever U^2 to overcome the resistance of the spring U′, or to be overcome by it. This lever, by means of the projection U^3, moves a cam sleeve, V, on the second-motion shaft Q′. The sleeve, although free to

slide along the shaft, rotates with it, and the cam surface cut on the sleeve acts on a roller so as to depress the steam-valve spindle R against the spring R′. The cam surface is so arranged that in one position of the sleeve V the steam-valve is held open during the whole revolution of the shaft Q′—that is, steam is admitted continuously by the steam-valve to the turbine. When, however, the solenoid gets more strongly

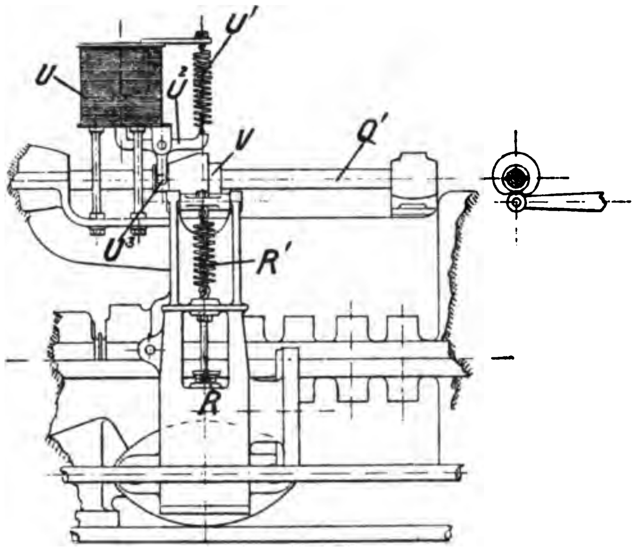


FIG. 151.—Electrical Governor for Parsons Turbine.

energized, it pulls the sleeve V to such a position that steam is admitted to the turbine only for a portion of a revolution of the shaft Q′; and, the greater the energizing current, the further the sleeve moves along, so that steam is admitted to the turbine for a smaller and smaller fraction of a revolution of the shaft Q′. The shaft Q′ is driven by the frictional contact of a wheel or disc carried by it with the end of the turbine spindle, the speed of revolution of the shaft Q′ being much less than that of the turbine spindle.

Figs. 152 and 153 show another arrangement of electric governor. Steam is admitted to the turbine by the double-beat valve A from the steam space A^1. Steam which leaks past the neck-bush C acts on the piston B^1 so as to force it upwards against the action of the spring B^2. This occurs when the valve D is closing the passages B^3 and B^4, but, when these passages are opened, the steam escapes from the lower end of the cylinder faster than it can enter by the leak; and so the piston B^1 descends, and, by means of the rod B, closes the main valve A. For intermediate positions of the valve D, the main valve assumes positions of partial opening. The eccentric G^1 is driven from the turbine spindle H by means of a worm and worm-wheel, and gives a rocking motion to the lever G. This is pivoted at G^2, and, consequently, its end E^1 has an up-and-down motion. This end, E^1, is connected to a lever, E, one end, E^2, of which is attached to the valve D. The lever E can turn about the point E^3, and the valve D will, therefore, be reciprocated up and down by the action of the eccentric G^1. This will allow regular puffs or blasts of steam to pass through the valve A.

The other end of the lever E is pivoted to the core F^1 of the solenoid F, which tends to draw it down against the action of a spring at E^3; so that an increase or diminution in the strength of the current energizing the solenoid will cause the lever E to turn about the point E^1 and actuate the valve D. The effect of the combined action of eccentric and solenoid is to prolong or shorten the duration of the puffs, and the turbine is thus governed. The number of puffs per minute is commonly from 100 to 200. With alternators a series coil may be employed in addition to the shunt solenoid in order to compound for constant voltage.

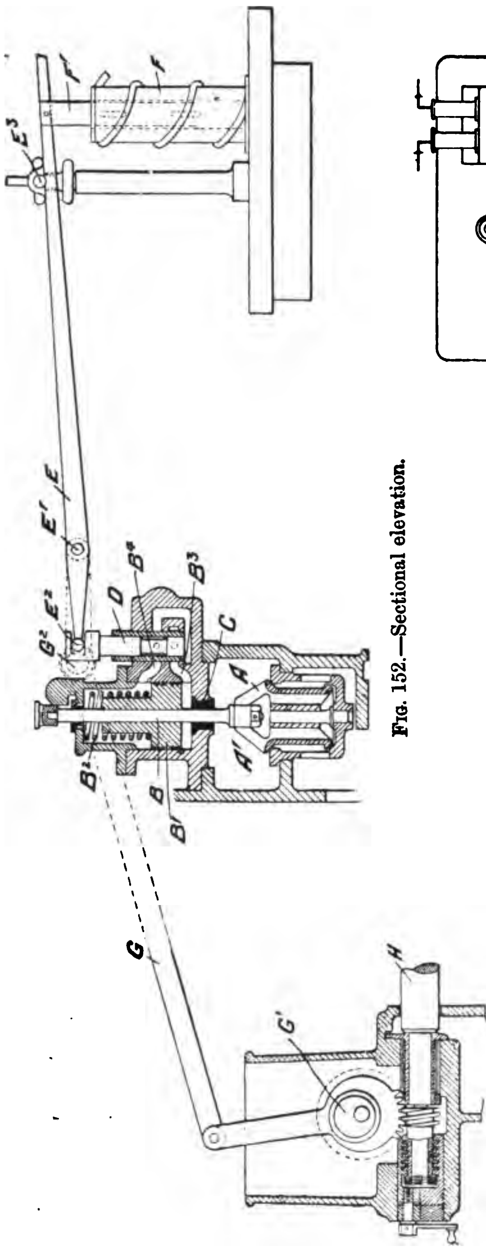


FIG. 152.—Sectional elevation.

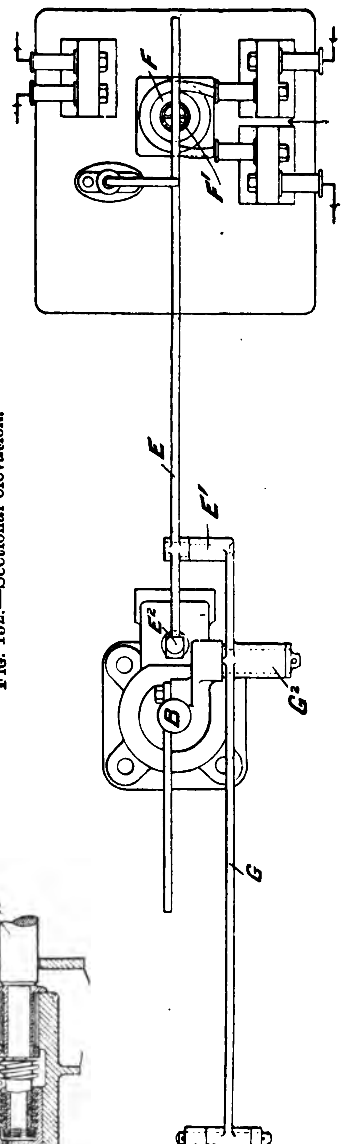


FIG. 153.—Plan.
Electrical Governor for Parsons Turbine.

The mechanism may be somewhat modified without departing from the principle of action. A slightly different mechanism is shown in Fig. 68, where P is the lever which is oscillated by the worm-driven eccentric Q, and which is adjustably connected to the lever R, which is pivoted at S to the lever T. This lever T corresponds to the lever E in Figs. 152 and 153, and has one end connected to the governor, while the other end actuates the valve. A centrifugal governor is shown in Fig. 68.

Centrifugal governors may be employed to control the admission of steam equally as well as electrical governors. For example, in Fig. 151 the sleeve V might have been actuated by a centrifugal governor mounted on and rotating with the second-motion shaft Q′. The electrical governor has the advantage in the driving of continuous-current electric generators when constant voltage is required, as it can control the voltage independently of the speed.

Superheating the steam before use very much improves the efficiency of a Parsons turbine. With moderate superheats a gain in efficiency of about 1 per cent. may be expected for every 8 or 9 degrees Fahr. of superheat up to at least 200 degrees Fahr. of superheat.

The bearings of a Parsons turbine are subjected to forced lubrication, the oil being supplied by a pump driven by an eccentric which receives its motion through the agency of a worm-wheel which gears with a worm on the turbine spindle. The same eccentric, worm and worm-wheel are commonly employed for the joint purpose of driving the oil-pump and of actuating the relay valve of the governor. The oil-pipes can be seen in Plate II. and in other illustrations. The oil is often passed through a tank cooled by circulating water, and used over and over again without leaving the machine.

Messrs. C. A. Parsons and Co. commonly construct their turbines with two steam-admission valves, through which the steam has to pass in series when it enters the turbine. An emergency or runaway governor is arranged to act on the first valve if the speed should become dangerous. Sometimes the emergency governor only releases a clip, and a weight closes the valve. In other cases the emergency governor opens a small valve, which allows the steam to escape from below the relay piston, which then falls and closes the steam-admission valve. The other valve is actuated by mechanism driven by the turbine spindle, and is controlled by a centrifugal or electrical governor, so that this valve opens or closes for longer or shorter periods in the manner just described, according to the load.

Parsons turbines are sometimes provided with a by-pass valve, which when open admits high-pressure steam to an intermediate portion of the turbine casing. This allows in an emergency full power to be obtained with a reduced steam-pressure (and with, of course, slightly reduced efficiency).

In order to allow for relative expansion between the turbine and armature spindles on the one hand and the fixed parts of a turbo-generator on the other hand, Messrs. C. A. Parsons and Co. connect their turbine and armature spindles together by means of a special clutch device which allows of a slight axial movement of the spindles relatively to each other. On the adjacent ends of the spindles are keyed discs having radially projecting teeth which enter slots cut in two flanged sleeves, which are then bolted together. The connection is rigid as regards angular movement, but a certain amount of axial play may take place. Provision is also allowed in supporting the turbine casing to allow it to expand when heated up. $\frac{5}{32}$ inch

of horizontal expansion has been measured on a 1800 K.W turbine between its two supports.

The glands where the spindle of a Parsons turbine leaves the casing contain metallic packing-rings arranged to cause very little friction. When a turbine is working condensing, live steam is supplied to the glands to prevent air leaking in and getting to the condenser. A little live steam is, of course, wasted, but this is inappreciable. When a turbine is exhausting into the atmosphere, an ejector fed with live steam is often used to eject from the gland the steam that would otherwise escape into the engine-room in what might be considered a slightly objectionable manner.

A 1000-kilowatt continuous current turbo-generator is shown in Figs. 154, 155, and 156 (Plate IX.), in front elevation, end elevation, and plan respectively. This machine was built by Messrs. C. A. Parsons and Co., and is now running at the Close Works of the Newcastle and District Electric Lighting Co. The steam enters the turbine first through an emergency or runaway valve, and then through a governor valve, controlled by a relay device similar to what has already been described, the arrangement of levers only being different. At the left of Fig. 154 can be seen the eccentric which actuates the relay valve, and also actuates the connecting rod of the oil-pump. There are two similar dynamos arranged tandem, the armatures being interchangeable. Either dynamo can be worked without the other. A claw-coupling is employed to connect together the turbine and armature shafts. A brush rocking-gear actuated automatically from the turbine keeps the brushes at the best positions on the commutators. This is accomplished by means of a piston, which is pressed upwards by the steam against the action of a spring. The machine is designed for

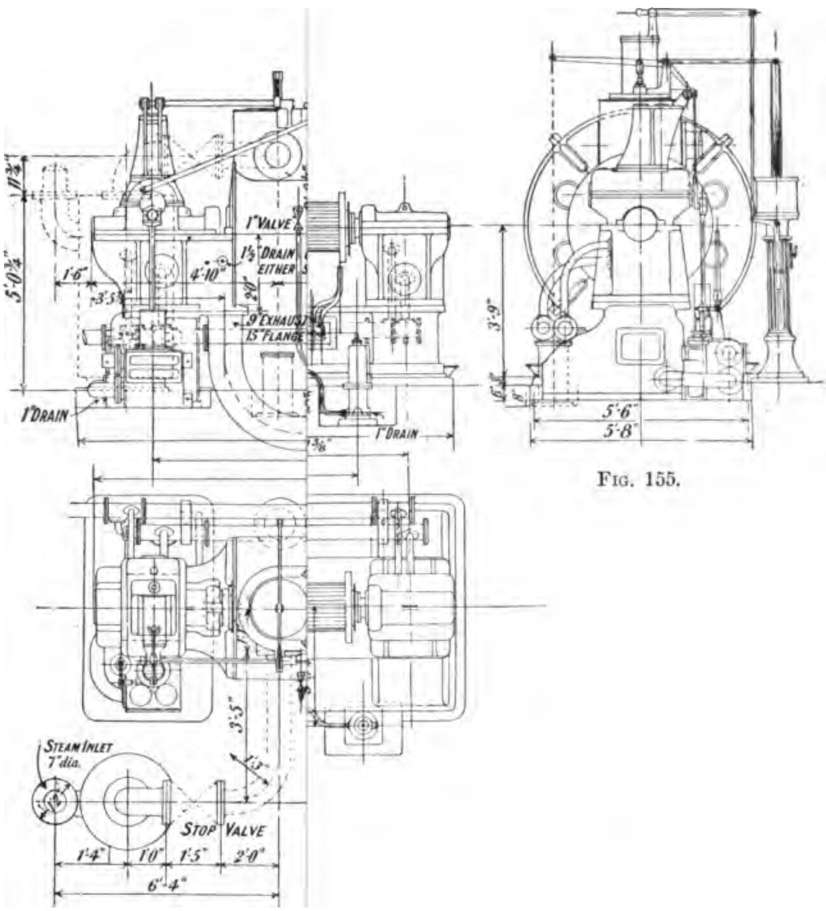

1" VALVE
1½" DRAIN
EITHER
4'-10"
5'-0¾"
11¾"
1'-6"
3'-3"
2'0"
9" EXHAUST
15" FLANGE
1" DRAIN
1" DRAIN
3'-9"
5'-6"
5'-8"
3'-5"
1'-3"
STEAM INLET
7" dia.
STOP VALVE
1'-4"
1'-0"
1'-5"
2'-0"
6'-4"

Fig. 155.

NE.

1800 revolutions per minute and 500 volts. The bottom part of the turbine casing weighs about 5¼ tons, and the top part about 3½ tons.

Plate I. shows one of two 1800-kilowatt turbo-generators constructed by Messrs. C. A. Parsons and Co. for the Dickinson Street electric power station of the Manchester Corporation. Figs. 157 and 158 (Plates X. and XI.) are respectively front elevation and plan of one of these machines. The steam enters by the breeches pipe F, and, after passing through the separator S, it enters the valve casing H, where it proceeds first through the emergency valve A, and then through the relay-operated valve B, which is of the nature of that described with reference to Figs. 152 and 153. D is the emergency governor which controls the closing-gear of the emergency valve A through the agency of the links, cranks, and shafts L,L. Should, through any cause or mishap, the speed exceed a certain amount, this emergency governor releases a catch and allows a weight to fall and close the valve. The governor E, which, like the governor D, is centrifugal, acts through mechanism on the link K, which is pivoted at its upper end to the lever M. The left-hand end of this lever is moved up and down by an eccentric through the agency of the rod N. The right-hand end of the lever M is provided with a weight O, and is suspended from the support P by means of a spring, the tension of which can be adjusted by hand. The combined movements of the rod N and the lever M actuate the relay-valve which controls the motion of the valve B.

Lubricating oil under pressure is supplied by a double-acting oil-pump. Water for keeping the bearings cool passes to the latter by way of the pipe Q, and is discharged by way of the pipe R.

Part of the exhaust pipe consists of a corrugated Fox tube T, which allows for expansion and contraction of the turbine relatively to the condenser. With the same object the high-pressure end of the turbine is not bolted rigidly to its stool, but is allowed to move longitudinally over the stool to which it is secured by bolts passing through elongated holes and provided with spring washers.

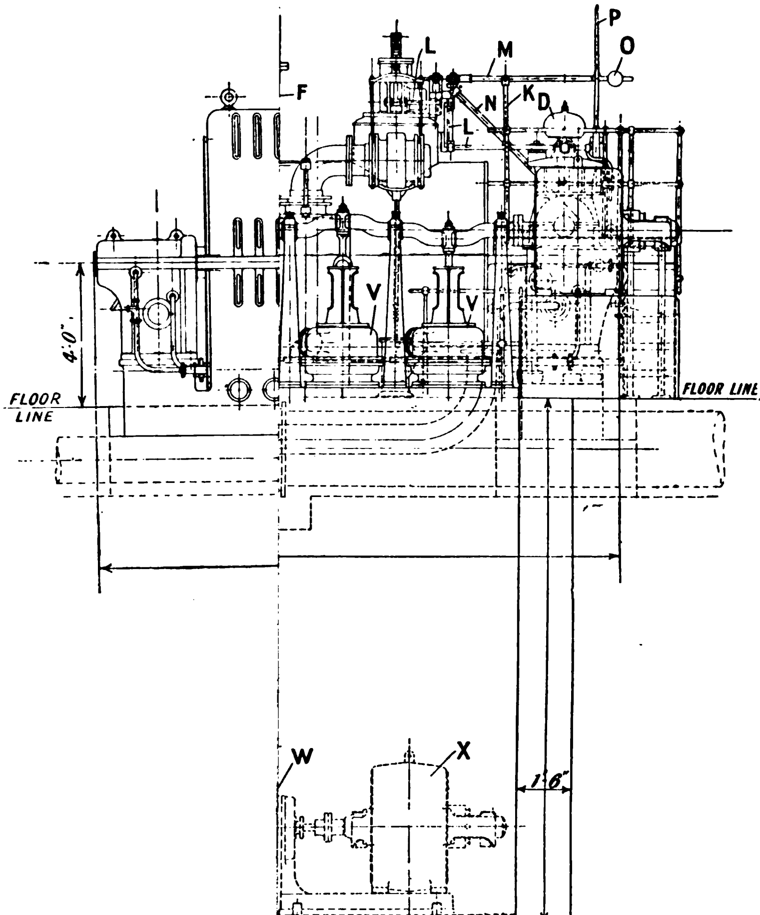
A jet condenser is employed, the jet-regulating hand-wheel being shown at U. The air-pump V, V, V, is of the three-crank type, and is driven by an electric motor. This pump removes air and vapour only, the water being removed by a centrifugal pump W driven by an electric motor X.

Provision is made for the turbine exhausting into the atmosphere by way of the pipe Y, which is connected to the exhaust end of the turbine by means of the casting Z, an automatic atmospheric exhaust valve being placed between Y and Z.

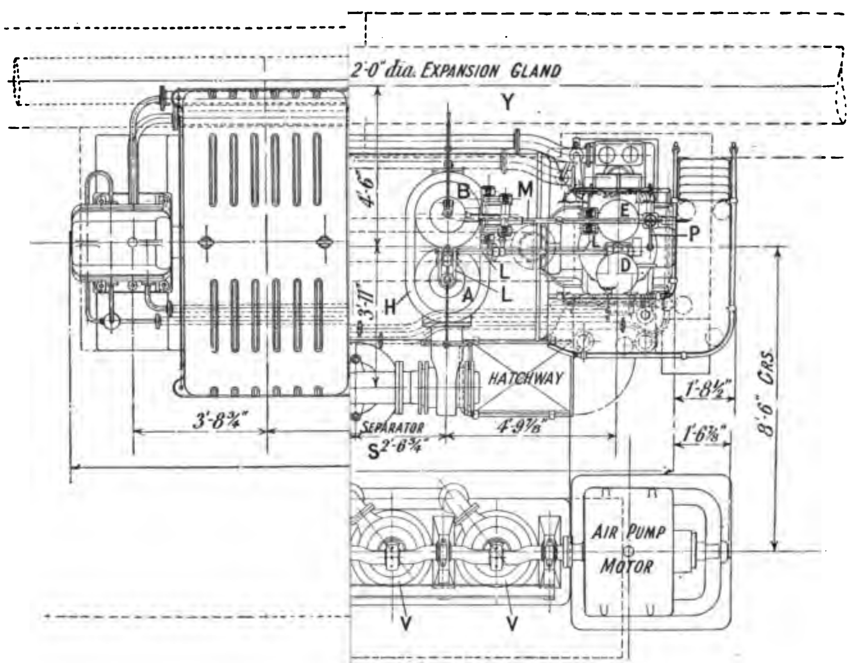
There are two dynamos; C, C are the commutators, the segments of which are held in place against the action of centrifugal force by means of the steel rings *r*. There are three of these rings on each commutator, one at the centre and one at each end. (Only the centre ones can be seen.) The brushes are automatically adjusted by steam gear, so as to be set forward at high loads and set back at low loads.

The machine runs at a little over 1000 revolutions per minute. Each dynamo has 6 poles, and the pressure is about 450 volts.

One of two 1250-kilowatt turbo-alternators supplied by Messrs. C. A. Parsons and Co. to the Corporation of Elberfeld, Germany, for the electric station of that city, is shown in Plate XVIII. The turbine has two cylinders—high and low pressure—arranged tandem. The alternator has four poles,



HESTER,


2'-0" dia. EXPANSION GLAND
Y
4'-6"
B
M
E
P
L
L
D
A
L
H
3'-11"
HATCHWAY
1'-8½"
8'-6" CRS.
3'-8¾"
SEPARATOR
S 2'-6¾"
4'-9⅞"
1'-6⅞"
AIR PUMP
MOTOR
V
V

IESTER.

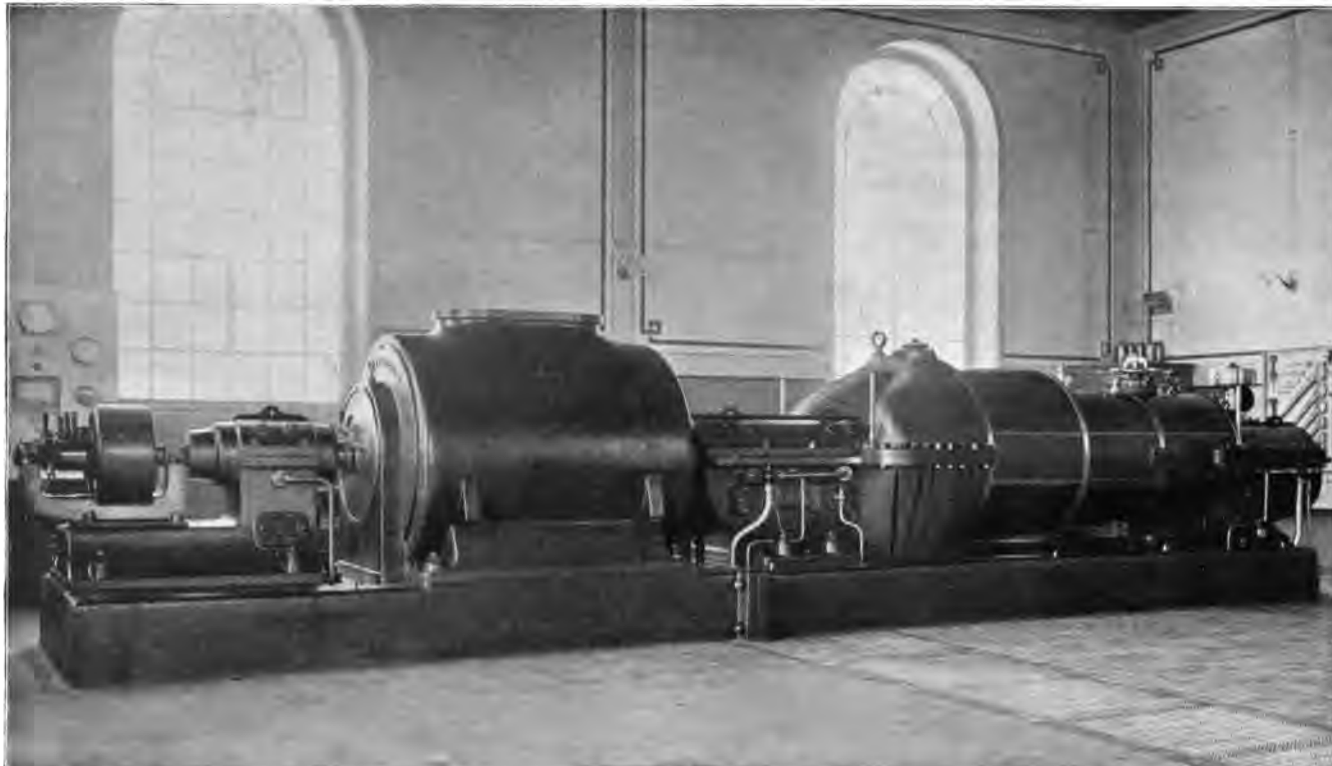


Plate XII.– 1200–1400 K.W. Turbo-Alternator constructed by Messrs. Brown, Boveri and Co.

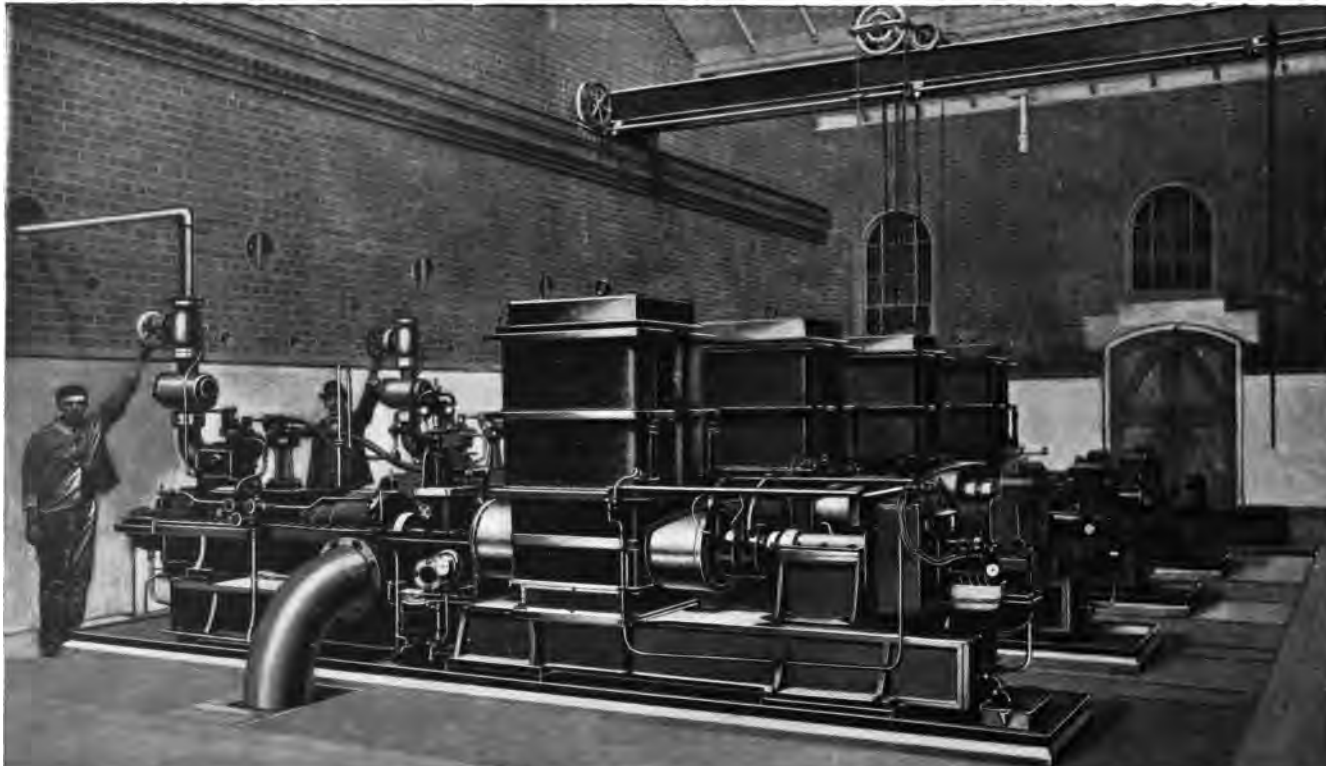


PLATE XIII.—VICTORIAN RAILWAYS LIGHTING STATION EQUIPPED WITH FOUR 150-KILOWATT PARSONS TURBINE ALTERNATORS WITH EXCITERS.

and supplies single-phase current at 4000 volts and 50 periods per second, the speed of rotation being 1500 revolutions per minute. The results of tests on this machine are given in Chapter XI.

Plate XII. shows a 1200–1400-kilowatt turbo-alternator constructed by Messrs. Brown, Boveri and Co. (of Baden, Switzerland), for the Power Transmission Works at Rheinfelden, Switzerland. At tests made on this machine at Rheinfelden in December, 1903, excellent results were obtained as regards steam consumption.

In Plate XIII. is seen the interior of the Victorian Railways Lighting Station. Four Parsons turbo-alternators, each of 150 kilowatts capacity, and provided with exciters, are there installed and run in parallel. Ferranti rectifiers are used, and the current employed for both arc and incandescent lighting.

Turbine-driven alternators may be built either with fixed armature and rotating fields, or with rotating armature and fixed fields. For high voltages the former arrangement is considered the better, and is usually adopted.

Parsons turbines have been very successfully applied to the driving of rotary pumps. Plate XIV. shows a steam turbine driving a centrifugal pump. This was supplied by Messrs. C. A. Parsons and Co. to Messrs. Storey Bros. and Co., of Lancaster.

The pump is normally employed for supplying water at a pressure of 22 lbs. per square inch to an ejector condenser. The steam for the turbine is then reduced by a throttle-valve from 60 lbs. per square inch to 20 lbs. per square inch. The speed of the turbine is not controlled in the usual way, but by the governor acting on another throttle-valve. The turbine and pump are also used as a reserve fire-engine in case the

regular fire-engine kept by Messrs. Storey should fail. When thus used the steam is admitted to the turbine at the full pressure of 60 lbs., and the governor put out of action. The water is then delivered at a pressure of 80 lbs. per square inch. Air is prevented from entering the pump-shaft glands by subjecting these to water pressure. For this purpose a water-tank is arranged above each gland, in which a constant head of water (several inches) is maintained, any surplus water overflowing into another tank, from which it is drained away.

Multiple-action pumps can be driven by steam turbines to deliver water at very high pressure. Two sets of high-pressure turbine pumps have lately been supplied by Messrs. C. A. Parsons and Co. to the Agent-General for New South Wales for use at the Sydney Waterworks. The first set comprises a steam turbine driving three high-speed centrifugal pumps. The three pumps working in parallel can raise 4½ million gallons of water every twenty-four hours to a height of 240 feet, and working in series they are capable of raising 1½ million gallons to a height of 720 feet. In the second set also one steam turbine drives three pumps. These in parallel can deliver 10 million gallons a day to a height of 80 feet, and in series 3⅓ million gallons a day to a height of 240 feet. In both sets a surface-condenser for the turbine is provided with the circulating water passages arranged as a by-pass to the main suction-pipe. Two air-pumps are provided, driven by worm gearing from the turbine spindle.

The first of these sets is illustrated in Plate XV., the three pumps being shown connected in series. In each case the water enters the pump by the lower port, and is discharged at the top opening. At a test of this machine made at Messrs. Parsons' works, water was discharged against as high a head

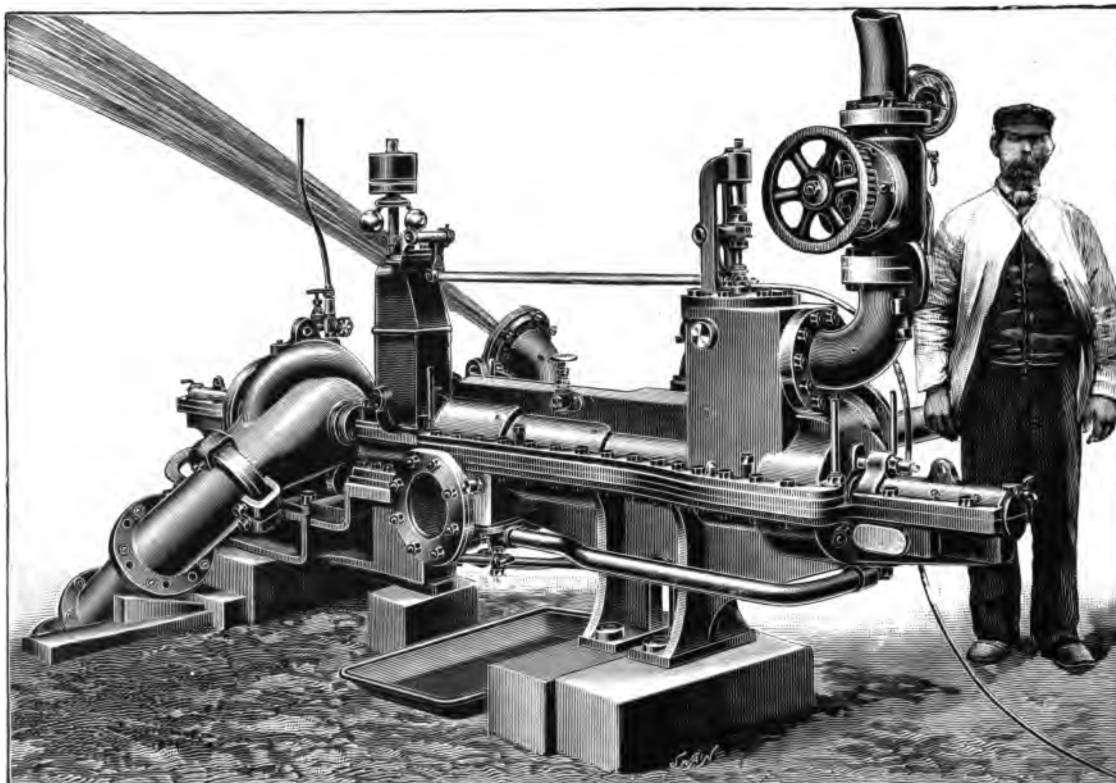


PLATE XIV.—PARSONS STEAM TURBINE COUPLED TO CENTRIFUGAL PUMP.

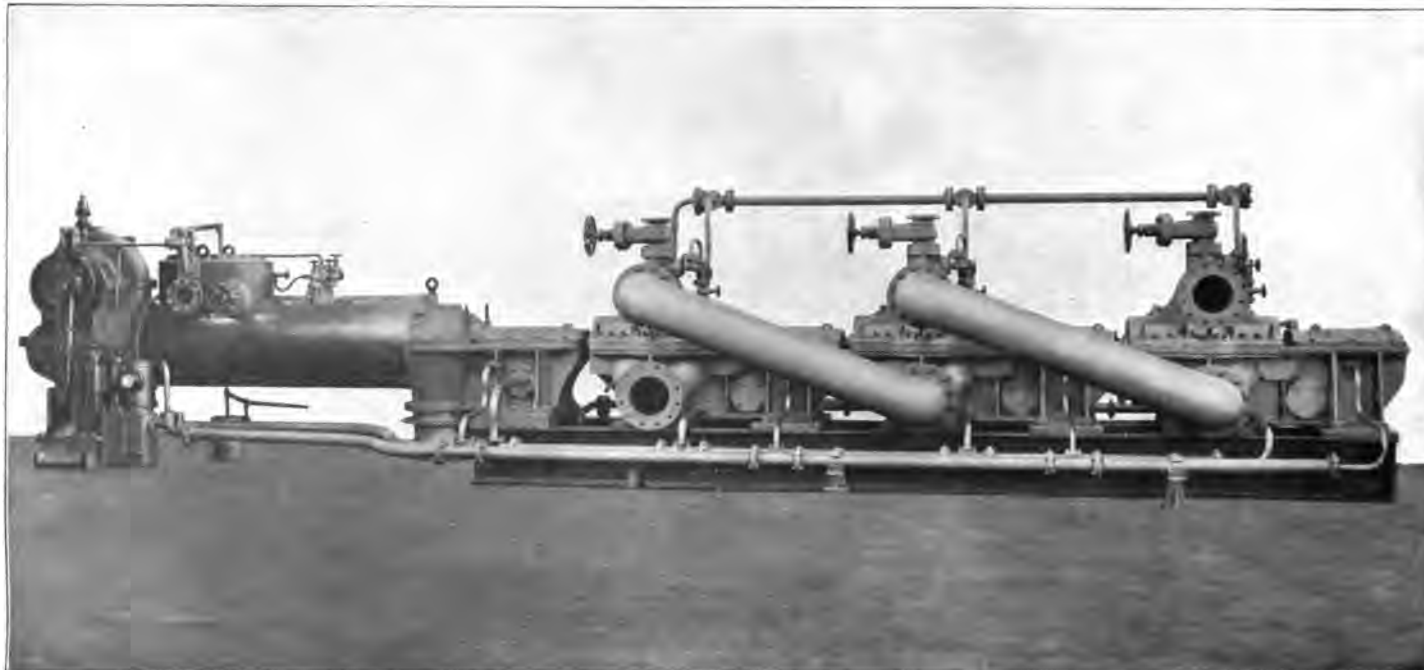


Plate XV.—Turbine-driven Pumps for Sydney Waterworks.

as 1000 feet, the speed being then 3700 revolutions per minute.

Parsons turbines are now largely employed to drive fans and air-propellers and compressors. The fans are usually of the screw-propeller type, and work in a cone. The high speed allows a large volume of air to be passed in proportion to the diameter of the fan. A 5-foot fan driven by a Parsons turbine installed at the Clara Pit, in Durham, can deliver 120,000 cubic feet of air per minute at 2-inch water gauge.

A turbo-fan is installed at Hulton Colliery, Chequerbent, near Bolton, and is used to ventilate by suction two mines which are worked as one as regards ventilation. A short wooden drift leads the air from the top of the upcast shaft to a cabin in which the turbine is situated. A conical iron casting 5 feet long has its larger end of 7 feet 1 inch diameter connected to the drift, and its smaller end of 3 feet 7 inch diameter connected to a short cylinder of about 3 feet 7 inch diameter, in which works the fan. An iron bend, through which passes the fan-shaft, conducts away the air from the fan to the discharge cone. The turbine, besides driving the fan, also drives a 50-kilowatt dynamo which supplies current for lighting and for actuating the motor which drives the circulating pump for the condenser. The motor and pump are situated in a separate building from the turbine. The fan-shaft is coupled to the low-pressure end of the turbine spindle, and the armature spindle to the high-pressure end of the same. The turbine, fan, and dynamo were constructed by Messrs. C. A. Parsons and Co. The pump for supplying the lubricating oil to the turbine, fan, and dynamo is driven from a shaft which is connected by a worm and worm-wheel to the turbine spindle, and rotates at $\frac{1}{40}$ of the speed of the turbine. This

shaft also, by means of an elliptical double-throw cam, actuates the relay valve for controlling the motion of the steam admission valve. The relay valve is conjointly controlled by a centrifugal governor. The exhaust pipe to the condenser is 18 inches in diameter, and the condenser is of the Ledward ejector type. Provision is also made for shutting off connection with the condenser, and exhausting into the atmosphere. The fan is 3 feet 6 inches in diameter, and has 8 blades. It rotates at about 3000 to 3500 revolutions per minute.

Table XVIII. gives the results of tests which were made on the turbine, fan, and dynamo by Mr. A. J. Tonge, of the Hulton Collieries.

TABLE XVIII.—TESTS OF TURBO-DYNAMO FAN AT HULTON COLLIERY.

Duration of trial in hours.	Steam pressure in lbs. per square inch.		Vacuum in inches of mercury.	Revolutions per minute of turbine, dynamo, and fan.	Air withdrawn by fan.			Electrical horse-power.	Total horse-power.	Percentage of full guaranteed load.	Steam consumption per total horse-power hour in lbs.
	At boiler.	Inside high-pressure end of turbine casing.			Quantity in cubic feet per minute.	Water-gauge.	Air horse-power.				
10¾	147	122	19¼	3360	112,000	5·18	91·3	51·6	143	100	32·2
13½	147½	122	19½	3320	109,300	4·95	85·2	56·0	141	99	32·6
18¾	147	125	19½	3200	97,500	6·10	93·7	54·8	148	104	29 2
8½	147	80	20½	3010	99,000	4·00	62·0	—	62	44	44·5

A Biram anemometer was used for the air measurements. Two-minute readings were taken, and the results averaged. The water-gauge pipe was perpendicular to the flow of the air and 10 feet from the fan. Readings taken at further distances from the fan were found to be the same.

It will be noted that the vacuum was not good. A much lower steam consumption would certainly have been obtained had the vacuum been better.

Fig. 159 shows the fan and its outside bearing, with the pipes for lubricating this bearing. The view is taken from the drift between the top of the upcast and the fan.

A fan driven by a Parsons steam turbine was run night and

FIG. 159.—Turbine-driven Ventilating Fan at Hulton Colliery.

day for five years at the Howdon Lead Works of Messrs. Cookson and Co., Wellington-Quay-on-Tyne. The fan was situated in the flue leading to the stack from the smelting furnaces, and had to maintain the draught required by four large calcining furnaces and two lead blast furnaces, all the gases being drawn through extensive condensing chambers for flue-dust. The water gauge close to the fan varied from

5 inches to 7 inches, this being necessary to overcome the resistance of the condensing chambers, and give the necessary draught at the furnaces. The fan is 3 feet in diameter, and ran at from 1400 to 2000 revolutions per minute according to requirements. Messrs. Cookson and Co. state that the fan gave no trouble during the whole time it was at work.

For delivering air at high pressures a compressor is employed which resembles a steam turbine, but of course is used as a generator instead of as a motor. Rings of fixed and moving blades are arranged alternately, and by these the pressure of the air is increased by steps. Two or more of these compressors can be arranged in series with or without intermediate coolers, and both or all can be coupled direct to the spindle of the actuating steam turbine. The stream of air delivered by such a compressor is, of course, continuous and uniform, and there are no suction and delivery valves continually opening and closing during action. In fact, no parts in the compressor have any motion except that of rotation, and as regards this perfect balance exists. By means of a by-pass valve which admits high-pressure steam to an intermediate part of the turbine casing, a higher speed can temporarily be obtained with consequent higher pressure of air-delivery.

A turbine compressor constructed by Messrs. C. A. Parsons and Co. is shown in elevation and plan in Figs. 160 and 161 (Plates XVI. and XVII.), respectively. The steam turbine is seen on the left, and the air-compressor with cooler on the right.

A compressor of this nature driven by a steam turbine and constructed by Messrs. C. A. Parsons and Co. for the Geo. Goch Mine, Johannesburg, has been built for an output of 4000 cubic feet of free air per minute at a pressure of 80 lbs., and at the time of going to press this compressor was undergoing its official tests.

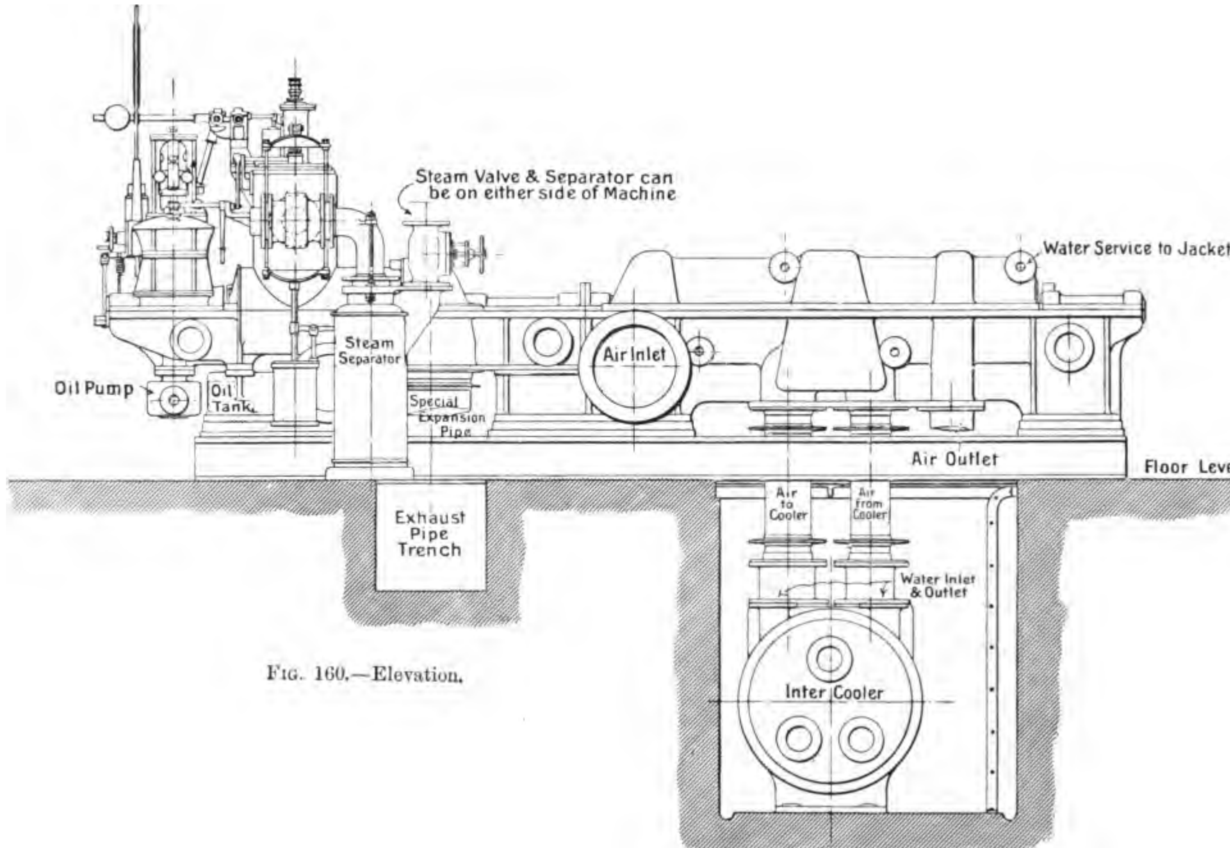


Fig. 160.—Elevation.

Plate XVI.—Turbo-Air-Compressor constructed by Messrs. C. A. Parsons and Co.

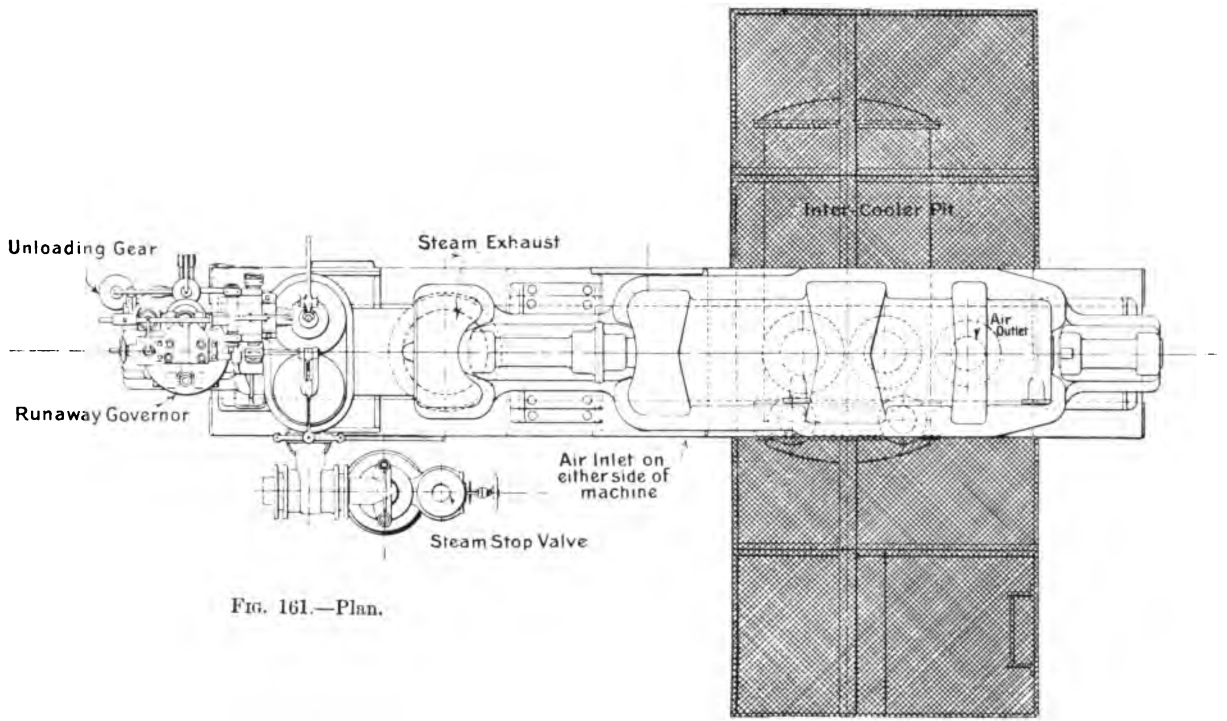


Fig. 161.—Plan.

Plate XVII.—Turbo-Air-Compressor constructed by Messrs. C. A. Parsons and Co.

CHAPTER XI.

SOME RECENT TESTS OF PARSONS TURBINES.

THIS chapter will be devoted to giving the results of some recent tests of Parsons turbines.

TABLE XIX.

TEST OF 24-KILOWATT TURBO-DYNAMO FOR MESSRS. SPILLERS AND BAKERS, NEWCASTLE-ON-TYNE, CONSTRUCTED BY MESSRS. C. A. PARSONS AND CO.

Pressure of steam above atmosphere at stop-valve.	Superheat at stop-valve.	Vacuum in the turbine cylinder. Bar.=30″.	Revolutions per minute.	Load.	Steam used.	
lbs. per sq. in.	degrees F.	ins. of mercury.		kilowatts.	lbs. per hr.	lbs. per kw.-hr.
80	0	28·8	4990	24·7	712	28·8
77	0	29·0	4630	11·8	400	33·9
74	0	29·1	4570	5·15	235	45·6
78	0	26·0	4900	23·8	798	33·5
79	0	0	4780	19·7	350	68·5

This shows that good efficiencies can be obtained even with steam turbines of comparatively small size. It also shows the effect of better vacuum and higher load on the steam consumption. The former was also shown by Tables XVI. and XVII., pp. 163 and 164.

TABLE XX.

50-KILOWATT STEAM ALTERNATOR SUPPLIED BY MESSRS. C. A. PARSONS AND CO. TO THE BLACKPOOL CORPORATION.

Pressure of steam above atmosphere at stop-valve.	Superheat at stop-valve.	Vacuum in the turbine cylinder. Bar.=30″.	Revolutions per minute.	Load.	Steam used.	
lbs. per sq. in.	degrees F.	ins. of mercury.		kilowatts.	lbs. per hr.	lbs. per kw.-hr.
126	0	28·0	5044	52·7	1480	28·0
132	0	28·5	4880	0	320	—

With a larger power and higher steam-pressure the efficiency here is slightly greater.

TABLE XXI.

TWO 100-KILOWATT CONTINUOUS-CURRENT TURBO-DYNAMOS FOR WEST BROMWICH ELECTRIC LIGHTING STATION, MADE BY MESSRS. C. A. PARSONS AND CO.

Pressure of steam above atmosphere at stop-valve.	Superheat at stop-valve.	Vacuum in the turbine cylinder. Bar.=30″.	Revolutions per minute.	Load.	Steam used.	
lbs. per sq. in.	degrees F.	ins. of mercury.		kilowatts.	lbs. per hr.	lbs. per kw.-hr.
129	54	27·8	3500	123	3144	25·5
134	64	27·7	3520	122	2913	23·8

With a greater power and moderate superheat the efficiency is again improved.

In January, 1901, a series of trials were made by Professor Ewing of a 500-kilowatt steam turbo-alternator at the works of the Cambridge Electric Supply Co.

The machine was constructed by Messrs. C. A. Parsons and Co., and erected at the Cambridge Electric Co.'s station in January, 1900, and ran at times daily, and at times intermittently, according to requirements, up to the time it was tested.

The turbine is of the parallel-flow type, with its shaft as usual directly coupled to the armature of the alternator, which is of the four-pole type, and designed to give 250 ampères at 2000 volts, running at 2700 revolutions a minute. The turbine is governed electrically, and is furnished with a surface condenser, and drives its own air-pump and circulating pump by means of a shaft carrying a screw-wheel driven by a worm on the main turbine shaft.

Table XXII. shows the collected results of the trials, and

Figs. 162 and 163 show the steam consumption graphically. The straightness of the line in Fig. 163 will be noticed.

TABLE XXII.

TESTS OF 500-KILOWATT PARSONS TURBO-ALTERNATOR AT CAMBRIDGE.

Trial number.		Effective electrical output in kilowatts.	Consumption of steam.	
			lbs. per hour.	lbs. per kw.-hour.
Trials of Jan. 9	1	518	12,970	25·0
	2	586	14,320	24·4
	3	273½	7,730	28·3
	4	160½	5,320	33·1
	5	0	1,850	—
Preliminary trials of Jan. 8	A.	535	13,350	25·0
	B.	300	8,270	27·6

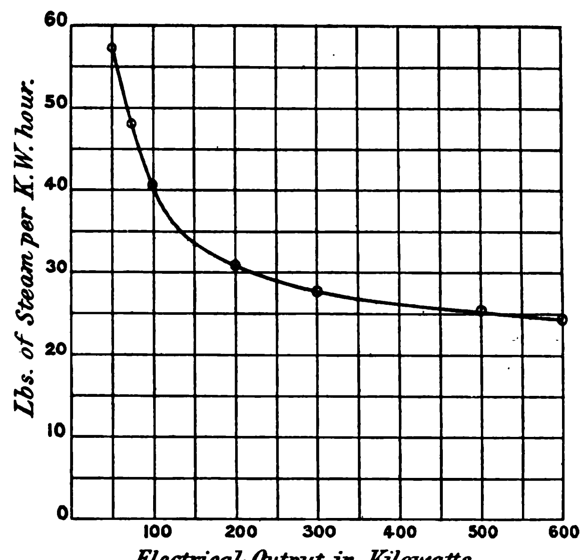


FIG. 162.—Steam Consumption of 500-Kilowatt Parsons Turbo-alternator at Cambridge.

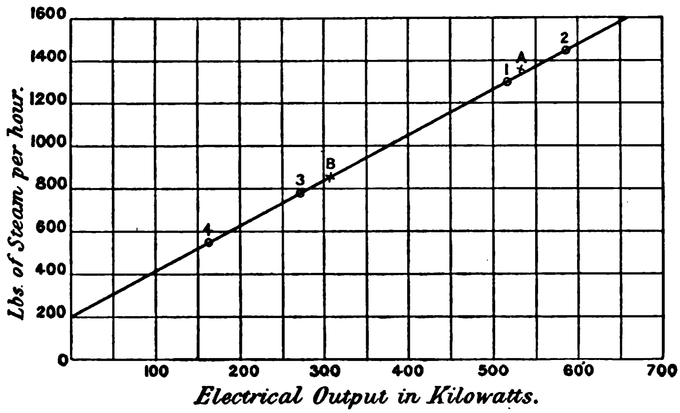


FIG. 163.—Steam Consumption of 500-Kilowatt Parsons Turbo-alternator at Cambridge.

Table XXIII. supplies particulars of the pressures, temperatures, speeds, etc.

TABLE XXIII.

TESTS OF 500-KILOWATT PARSONS TURBO-ALTERNATOR AT CAMBRIDGE.

Number of trial.	1.	2.	3.	4.	5.	A.	B.
Electrical output in kws.	518	586	273½	160½	—	535	300
Volts at terminals of generator	2,100	2,150	2,250	2,290	2,280	2,120	2,110
Speed in revolutions per minute	2,670	2,740	2,630	2,590	2,580	2,880	2,800
Air-pump discharge, lbs. per hour	12,970	14,320	7,730	5,320	1,850	13,350	8,270
Air-pump discharge, lbs. per kws. per hour	25·0	24·4	28·3	33·1	—	25·0	27·6
Pressure at stop-valve, lbs. per sq. in.	148	145	151	151	121	145	150
Vacuum in condenser, inches	27·8	27·9	28·2	28·3	28·3	26·6	27·6
Vacuum in turbine cylinder, inches	25·7	25·4	27·2	27·8	28·1	25·1	26·2
Temperature of air-pump discharge, ° F.	74	76	57·5	56	54	90	68
Temperature of circulating water, inlet, ° F.	40	40	38	39	36	41	39
Temperature of circulating water, outlet, ° F.	71	72·5	60	57	46	91	71
Barometer, inches ...	29·93					29·99	

In January, 1900, tests were made at the works of Messrs. C. A. Parsons and Co., Newcastle-on-Tyne, of a 1250-kilowatt

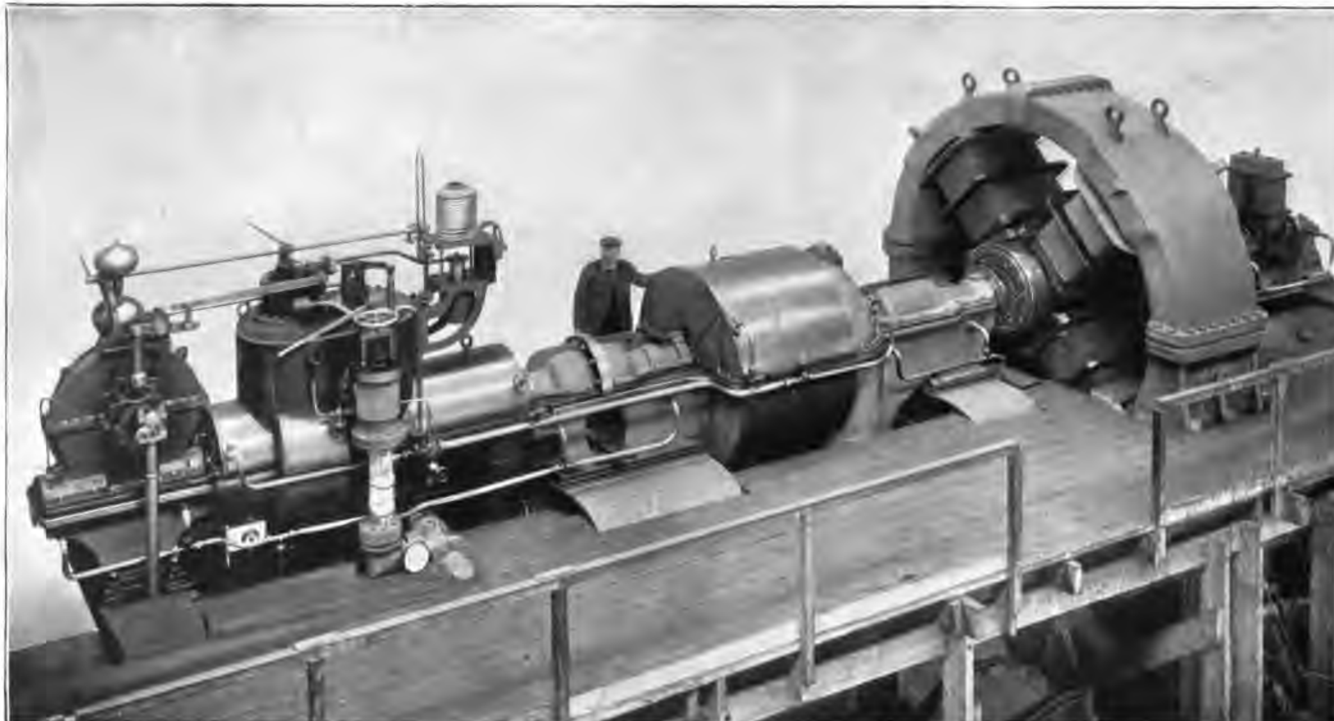


PLATE XVIII.—1250-KILOWATT PARSONS TURBO-ALTERNATOR FOR ELBERFELD CORPORATION, AS ERECTED FOR THE TRIALS AT THE HEATON WORKS, NEWCASTLE-ON-TYNE.

turbine-generator, constructed by that firm for the city of Elberfeld. This machine is shown in Plate XVIII., and was referred to on p. 178. The tests were conducted by W. H. Lindley, Esq., M.Inst.C.E., and Professors Schröter and Weber of the Polytechnicum, Zurich. Steam was supplied by one Babcock and Wilcox boiler, two marine boilers, and a locomotive boiler. A Babcock and Wilcox superheater with independent firing was introduced into the main steam-pipe. The machine was loaded with a water resistance consisting of four electrodes immersed in four iron vessels fitted with water coolers, while an auxiliary adjustable water resistance was employed to regulate the load.

The tests extended over three days, exclusive of a preliminary trial, and the results as regards steam consumption are given in Table XXIV.

TABLE XXIV.

TESTS OF 1250-KILOWATT PARSONS STEAM TURBO-ALTERNATOR FOR ELBERFELD CORPORATION.

Number of series.	Amount of load.	Exact value in output in kws.	Steam consumption per kw.-hour.		Steam consumption in one hour.
			lbs.	kgs.	kgs.
A.	Preliminary trial	1172·7	18·22	8·26	9,689
II.	Overload	1190·1	19·43	8·81	10,485
I.	Normal load	994·8	20·15	9·14	9,092
III.	Three-quarter load	745·3	22·31	10·12	7,542
IV.	Half load	498·7	25·20	11·42	5,695
V.	Quarter load	246·5	33·76	15·31	3,774
VI.	No load with alternator excited	0	—	—	1,844
VII.	No load without excitation ...	0	—	—	1,183

The same steam-pressure and the same amount of superheat were not used in all the trials. The steam consumption was, therefore, calculated by the experts conducting the tests for a steam temperature of 197·3° C., this being a superheat of

14·3° C.; and, to enable a comparison to be made with the steam consumptions of engines working with saturated steam, the equivalent consumptions for saturated steam at eleven atmospheres were calculated. The results are given in Table XXV.

Fig. 164 shows the steam consumption graphically.

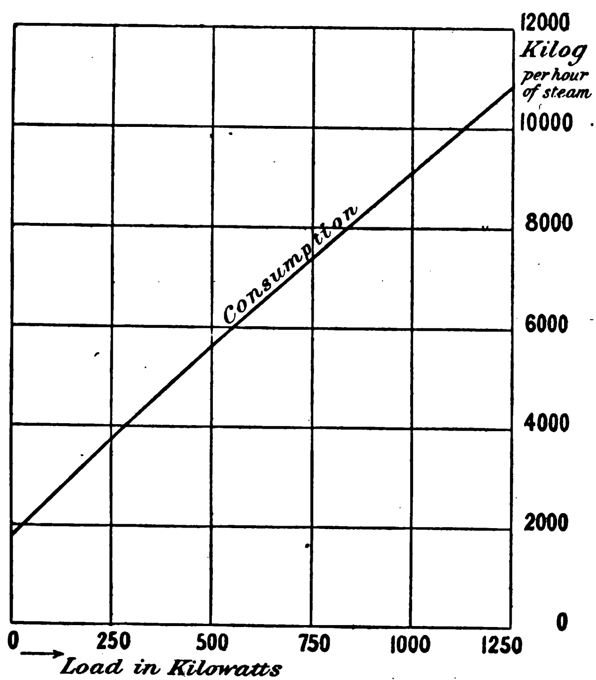


FIG. 164.—1250-Kilowatt Parsons Turbo-alternator. Diagram of total steam consumption per hour.

Table XXVI. shows the variation in the speed between no load and full load. The number of revolutions per minute was obtained by noting the time occupied by 200 revolutions of the driving-wheel of the valve-gear and air-pump, this driving-wheel rotating at one-eighth of the speed of the turbine.

TABLE XXV.

Tests of 1000-Kilowatt Parsons Turbo-alternator.

Number of series.	Load in kilowatts.	Average observed steam pressure in $\frac{kgs.}{cm.^2}$ absolute.	Corresponding temperature of saturated steam.	Average observed temperature of superheated steam at inlet valve.	Superheating (Col. 5 – Col. 4).	Observed steam consumption per kw.-hour.	Total heat contained in 1 kg. of steam at observed steam pressure. In saturated condition.	Total heat contained in 1 kg. of steam at observed steam pressure. In superheated condition.	Measured consumption of heat per kw.-hour (Col. 9 × Col. 7).	Corresponding consumption of saturated steam per kw.-hour (Col. 10 ÷ Col. 8).	Corresponding consumption of steam at 11 kgs. absolute and at 14·3° superheating (1 kg. of steam = 669·2 cal.).	Corresponding consumption of saturated steam at 11 atms. absolute (1 kg. of steam = 662·3 cal.).
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)
	kilowatts.	$\frac{kgs.}{cm.^2}$	° C.	° C.	° C.	kgs.	calories.	calories.	calories.	kgs.	kgs.	kgs.
II.	1190·1	10·11	179·3	189·5	10·2	8·81	661·1	666·0	5,867	8·87	**8·76**	8·86
I.	994·8	10·47	180·9	192·0	11·1	9·14	661·7	667·0	6,096	9·21	**9·11**	9·20
III.	745·3	10·76	182·0	190·0	8·0	10·12	662·0	665·8	6,738	10·18	**10·07**	10·17
IV.	498·7	10·40	180·6	209·7	29·1	11·42	661·6	675·6	7,715	11·66	**11·53**	11·66
V.	246·5	10·14	179·4	196·4	17·0	15·31	661·2	669·4	10,248	15·50	**15·31**	15·47
VI.	No load with excitation.	10·34	180·3	193·0	13·3	per hour. 1844	661·5	667·8	per hour. 1,231,423	per hour. 1861	per hour. **1840**	per hour. 1859
VII.	No load without excitation.	10·49	181·0	194·5	13·5	1183	661·7	668·2	790,481	1194	**1181**	1194

TABLE XXVI.

1000-KILOWATT PARSONS TURBO-ALTERNATOR. VARIATION IN SPEED BETWEEN NO LOAD AND FULL LOAD.

Time.	Load.	Steam pressure.	Vacuum in condenser.	Potential of alternator.	No. of revolutions as counted.		Variation in the number of revolutions.	Variation per cent.
					No load.	Full load.		
h. m.	kws.	lbs.	mm.	volts.				
10 44-45	0	150	—	3705	1482	—	—	—
11 16-17	1020	140	693	3960	—	(1433)	(−49)	(3·3)
0 19-20	1035	140	691	3950	—	1424	−58	3·9
11 28-30	0	150	712	3900	1486	—	+62	4·3
11 35-36	1040	145	696	4060	—	1429	−57	3·8
11 44	0	140	712	3880	1472	—	+43	3·0
0 48	960	140	698	4045	—	(1433)	(−39)	(2·6)
0 52	1058	140	693	4040	—	1429	−43	2·9
—	—	—	—	Average	1480	1427	53	3·6

Fig. 165 shows the effect on the speed of governing with a centrifugal governor with an increasing load, while Fig. 166

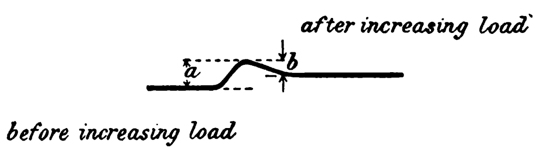


FIG. 165.—Increasing Load.

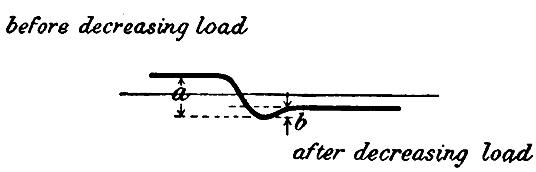


FIG. 166.—Decreasing Load.
Variation in speed with centrifugal governor.

shows the same with a decreasing load. Table XXVII. gives a summary of the results, the numbers in the fifth, sixth, and seventh columns referring to the distances marked on the diagrams (Figs. 165 and 166).

TABLE XXVII.

Test.	Average of all values of load.	Limits: Variation in the load			Variation in speed.			Variation in potential.			Average of variations in the load. Kilowatts.
		within the limit of	in per cent.		Average.			In per cent.		Average.	
	kilowatts.	kilowatts.	max.	min.	a	b	a–b	+	–	volts.	From ⇄ To
IXa	957	1086–840	19·5	16·3	1·75	0·67	1·08	1·29	1·20	50	1050 ⇄ 864
IXb	694	790–590	26·7	16·4	1·28	0·65	0·63	1·19	1·35	51	766 ⇄ 623
IXc	497	590–400	47·5	30·5	1·36	0·73	0·63	1·32	1·28	53	590 ⇄ 404
IXd	405	500–306	63·4	36·0	1·62	0·86	0·75	1·35	1·41	56	490 ⇄ 312
IXe	251	292–204	43·1	26·9	1·37	0·63	0·74	1·34	1·29	51	292 ⇄ 210 and back.

As the average potential may be taken at 4000 volts, the actual variation of 52 volts on the average amounts to **1·3** per cent. of the initial potential.

TABLE XXVIII.

Test.	Average of all values of load.	Limits: Variation in the load			Speed.					Variation in voltage.			Average of variations in the load. Kilowatts.
		within the limits of	in per cent.		Variations.			Average. (a–b)		In per cent.		Volts.	
	kilowatts.	kilowatts.	max.	min.	a	b	d	+	–	+	–		From ⇄ To
Xa	281	336–222	51·3	27·5	—	—	—	0·158	0·227	1·05	1·10	43	230 ⇄ 332
Xb	492	616–380	62·1	34·4	0·31	0·22	1·32	0·84	0·75	1·10	1·15	45	382 ⇄ 601
Xc	714	(900)–580	55·2	12·2	0·24	0·20	1·29	0·99	0·73	1·11	1·10	44	611 ⇄ 818
Xd	900	1016–790	30·6	19·3	0·21	0·27	1·26	0·86	0·80	1·06	1·12	45	797 ⇄ 1007 and back.

The average variation in the potential amounts to 44 volts, *i.e.* to **1·1** per cent. of the initial voltage.

Figs. 167 and 168 and Table XXVIII. show the effects of governing with an electrical governor.

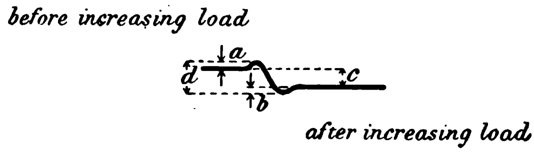


FIG. 167.—Increasing Load.

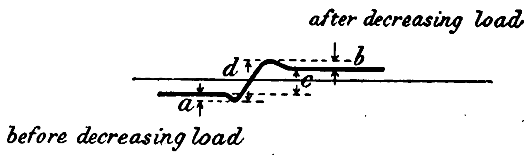


FIG. 168.—Decreasing Load.
Variation in speed with electrical governor.

It will be noticed that the centrifugal governor increases the speed with diminishing load and reduces the speed with increasing load, while the action of the electrical governor is the reverse.

The low steam consumption obtained with the Elberfeld machine has since been beaten by more than one large turbine constructed by Messrs. C. A. Parsons and Co.

CHAPTER XII.

SOME OTHER STEAM TURBINES.

Westinghouse-Parsons Steam Turbines, built by the Westinghouse Machine Company of Pittsburg, U.S.A., and by the British Westinghouse Electric & Manufacturing Company, Limited, of Manchester, are of the same nature as the turbines built by Messrs. C. A. Parsons & Co.; but it is convenient to consider them here.

In 1896 the Westinghouse Machine Company acquired the rights to manufacture Parsons steam turbines in the United States, and since then have built a considerable number. Figs. 169, 170, and 171 show a 500-H.P. steam turbine constructed by this company and coupled to a 300-K.W. bipolar, two-phase alternator. The author is indebted for these views to an article by Mr. B. Harding in the *Engineering Record,* New York. The steam inlet can be seen in the plan and end elevation, and the governor-valve close beside it is seen in the former figure. The governor is situated over the low-pressure end of the turbine. The oil-pump for supplying lubricant under pressure can also be seen at this end; it is driven by a worm-wheel gearing with a worm situated at the coupling between the turbine and alternator shafts. The turbine is of the parallel-flow type, and the steam in its passage along the widening cylinder from the high-pressure end to the

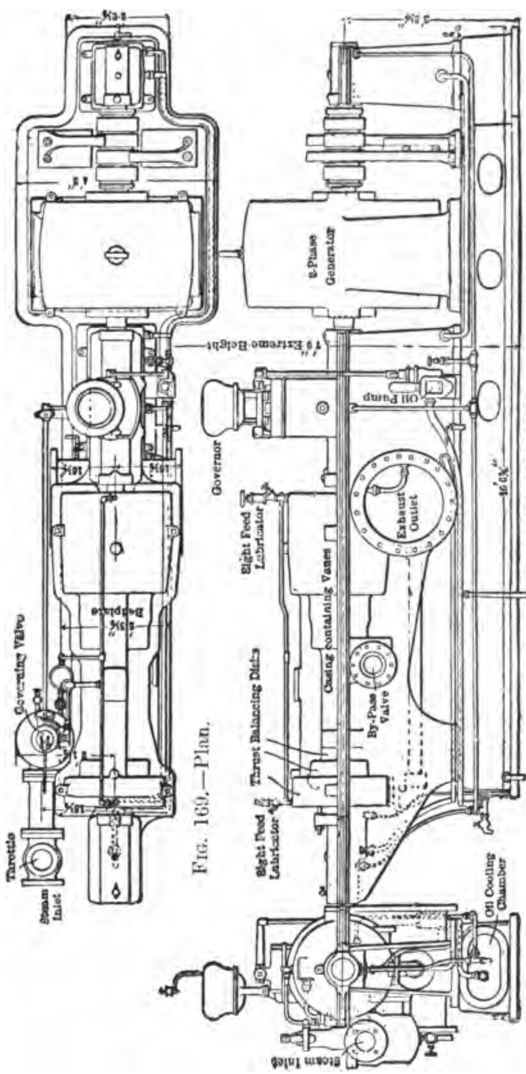


FIG. 169.—Plan.

FIG. 170.—End Elevation.

FIG. 171.—Side Elevation.

exhaust end acts in succession on 58 rings of blades, and is expanded about ninety-six-fold, that is from 120 to 125 lbs. pressure above atmosphere down to 1 to 1½ lbs. abs. Each blade contributes at full load to the extent of about 1 oz. to the force required to turn the turbine spindle. The speed is 3600 revolutions per minute, the voltage 440, and the alternations per second 120.

The turbine and alternator are carried on the same bed-plate, and this rests without holding-down bolts on brick piers, which are merely sufficient to carry the weight of the machine. The complete machine—turbine, alternator, and bed-plate—weighs about 25,000 lbs. The principal dimensions are given in the figures.

Plate XIX. illustrates the engine-room of the Westinghouse Air Brake Company at Wilmerding, Pa., U.S.A., in which four of these turbo-alternators are installed, and supply electric energy for light and power.

A by-pass valve is provided on each turbine, and this is used when the turbine has to work at overload or when the boiler-pressure is reduced or the condenser vacuum impaired. The by-pass valve admits high-pressure steam to an intermediate part of the turbine casing. This, of course, reduces the efficiency. The effect of opening the valve can be well seen in Fig. 171A. Here the upper curve indicates the steam consumption of the turbine when run non-condensing. The by-pass valve has been opened at a little over 200 E.H.P. The lower curve shows the steam-consumption of the turbine when exhausting into a condenser.

A test was recently made by Professor Robb, of the Electrical Engineering Rensselaer Polytechnic Institute, on a Westinghouse-Parsons steam turbine constructed by the

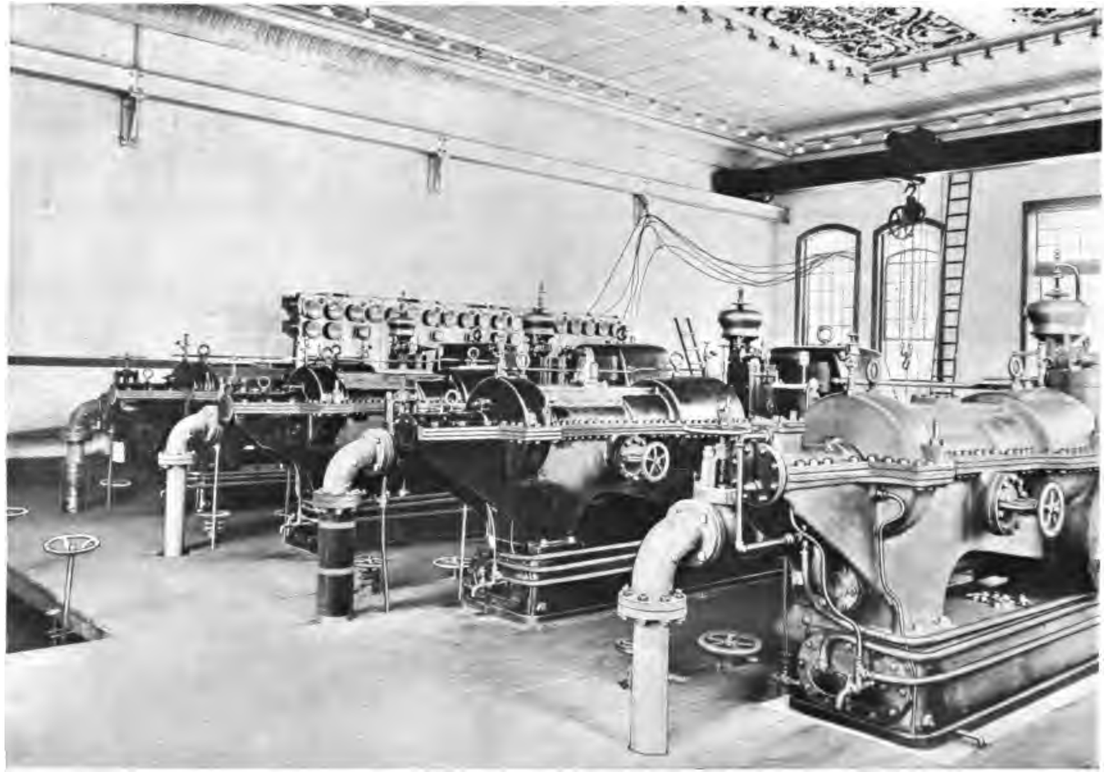


PLATE XIX.—ENGINE-ROOM OF THE WESTINGHOUSE AIR BRAKE COMPANY, WILMERDING, PA., U.S.A., CONTAINING FOUR 500 H.P. WESTINGHOUSE-PARSONS STEAM TURBINES.

Plate XX.—Westinghouse-Parsons Steam Turbine driving 1500-Kilowatt Alternator, Pearl Street Station, Hartford Electric Light Company, U.S.A.

Westinghouse Machine Company, and installed at the Pearl Street Station of the Hartford Electric Light Company at Hartford, Conn., U.S.A. This turbine, with the two-phase alternator which it drives, is shown in Plate XX. The machine is rated at 1500 kilowatts, but is capable of working at 2000 kilowatts for

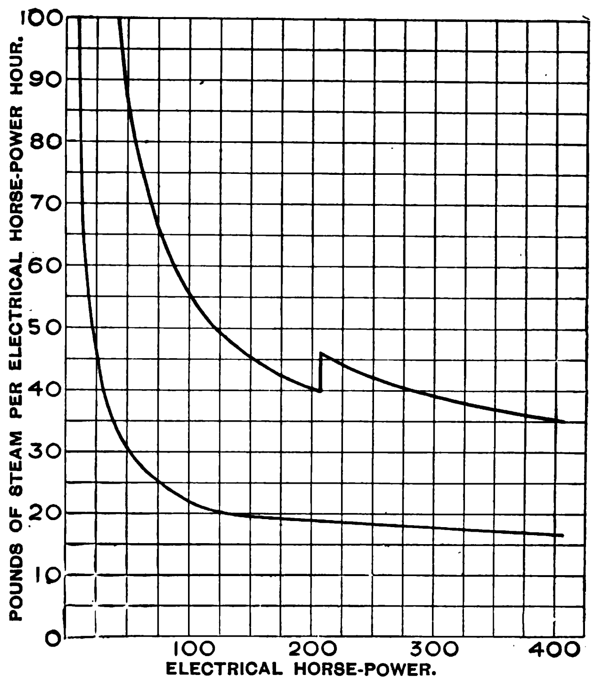


FIG. 171A.—Steam Consumption Curves of Westinghouse-Parsons Steam Turbine.

several hours, as can be seen in Table XXIX., which shows the results of the tests made by Professor Robb. These tests were made under ordinary station conditions. The steam consumption tabulated is that of the turbine only and not of the auxiliaries.

There are in all in this turbine about 30,000 fixed and moving

TABLE XXIX.

Tests of Turbo-generator at Hartford, Conn., U.S.A.

Test.		Load.			Length of test.	Steam, mean gauge pressure.	Barometer.	Vacuum at turbine.	Superheat. Degrees Fah.			Steam consumption.	
No.	Date. 1902.	Average K.W.	Max. K.W.	Min. K.W.	Hours.	lbs. per square inch.	Inches.	Inches.	Mean.	Max.	Min.	lbs. per E.H.P. hour.	lbs. per K.W. hour.
1	Jan. 27	748	885	580	6	155·5	30·70	26·22	0	0	0	24·13	32·17
2	Jan. 28	1657	1820	1480	6	151·3	30·73	28·00	40·08	61·34	19·85	15·15	20·2
3	Feb. 1	1998	2185	1900	4	155·4	30·27	26·91	41·56	55·05	32·45	14·43	19·10
4	May 7	471	730	310	6	151·8	29·86	26·62	19·10	29·00	3·50	23·97	31·96
5	May 8	888	980	750	6	152 6	30·04	25·83	32·90	47·50	12·00	19·90	26·53
6	May 9	1371	1570	1110	6	151·9	29·81	26·26	32·10	38·60	12·50	16·46	21·94
7	May 12	834	940	660	6	153·2	30·26	27·26	35·40	45·10	20·10	18·50	24·60
8	May 13	364	520	150	6	153·1	30·06	27·40	29·0	41·00	2·50	25·10	33·47

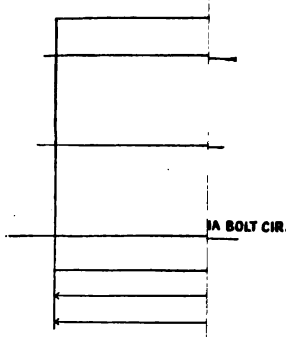


Fig. 175.

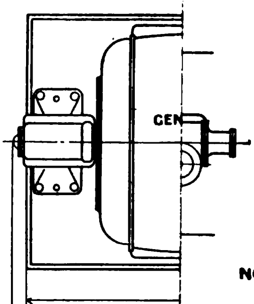


Fig. 172.

NOTE:- NO FOUNDATION BOLTS REQUIRED.

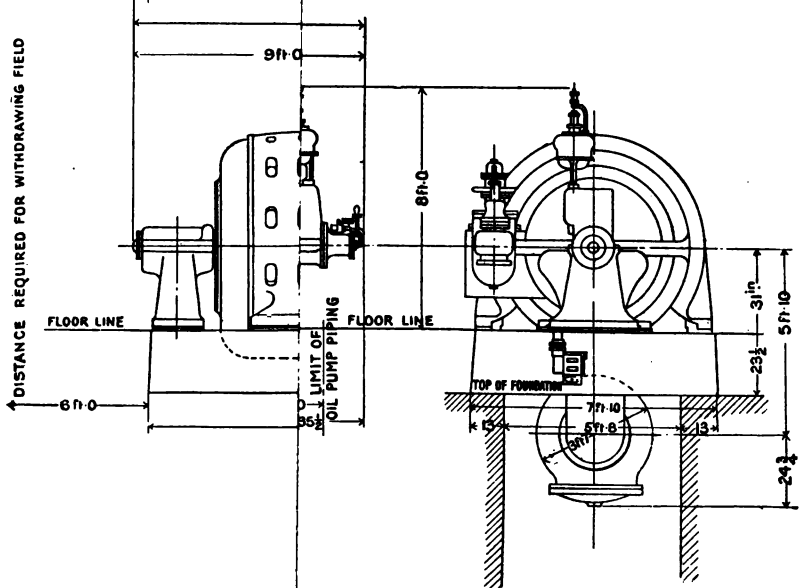


Fig. 174,

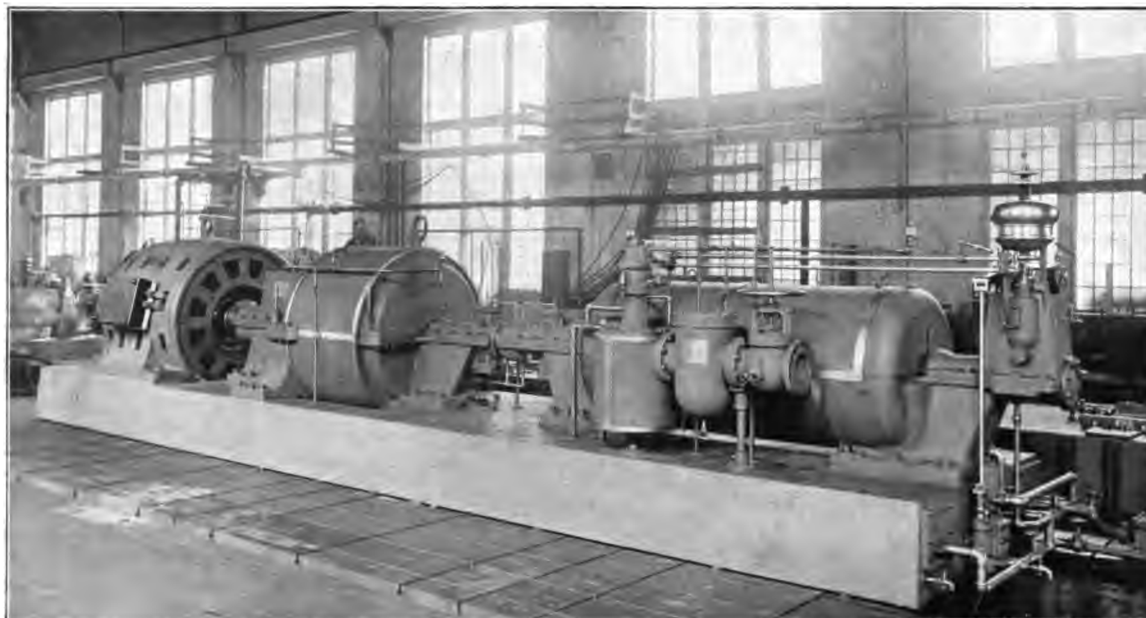


PLATE XXII.—750-KILOWATT WESTINGHOUSE-PARSONS COMPOUND STEAM TURBINE AND GENERATOR.

blades, varying in length from $1\frac{3}{4}$ inches at the high-pressure end to 8 inches at the low-pressure end. The governor is actuated from a worm on the extreme (high-pressure) end of the turbine spindle. This worm is also used to actuate the pump for the lubricating oil. A by-pass valve is provided as on the 500-H.P. machine already described to admit high-pressure steam when desired to an intermediate part in the turbine casing. The exhaust outlet from the turbine is about 10 square feet in section. The length of the turbine and alternator complete is 33 feet 3 inches, and its breadth is 8 feet 9 inches. The total weight of turbine and alternator in running order is about 175,000 lbs., which (taking the power of the machine at 2000 K.W.) works out at $87\frac{1}{2}$ lbs. per K.W. The length of the rotating part of the turbine only is 19 feet 8 inches over all, its greatest diameter is 6 feet, and its weight 28,000 lbs. The distance between the bearings of the turbine is 12 feet 3 inches.

Plate XXII. shows a 750-K.W. Westinghouse-Parsons compound turbine and generator. The high-pressure cylinder is seen at the right, with the throttle-valve, strainer, admission-valve, and governor gear. The generator can be seen at the left of the plate.

Figs. 172, 173, and 174 (Plate XXI.) show in plan, front elevation and end elevation respectively, a 1250-K.W. compound condensing steam turbine and six-pole, three-phase, revolving-field alternator constructed by the Westinghouse Machine Company for the Rapid Transit Subway at New York. Fig. 175 (Plate XXI.) gives the foundation and steam inlet and exhaust outlet dimensions. The revolutions per minute are 1200, which give 120 alternations per second. The voltage is 11,000.

Some large turbo-alternators at present being constructed by the British Westinghouse Electric and Manufacturing Company, Limited, rated at 5500 kilowatts, but expected to be capable of a 50 per cent. overload, have an over-all length of 51 feet 9 inches, an extreme breadth of 14 feet, and an extreme height of 12 feet. The length of each turbine alone is 29 feet. These machines are designed for a steam pressure of 165 lbs. per square inch and for a speed of 1000 revolutions per minute.

The Stumpf Steam Turbine, the invention of Professor Stumpf of Berlin, is of the same nature as the De Laval turbine —that is, the expansion of the steam is completed before the fluid enters the

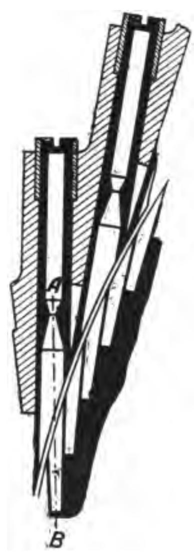


FIG. 176.—Nozzles and Buckets of Stumpf Steam Turbine.

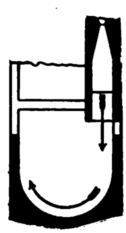


FIG. 177. — Single-bucket arrangement of Stumpf Steam Turbine.

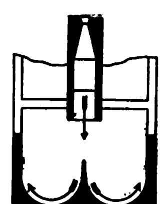


FIG. 178. — Double-bucket arrangement of Stumpf Steam Turbine.

turbine buckets. In the Stumpf turbine, however, the nozzles are in the plane of the wheel instead of making an angle with this plane as in the De Laval turbine. Fig. 176 shows part of the rim of a Stumpf turbine wheel and a couple of nozzles. A section of a bucket on the line AB of Fig. 176 may be as shown in Fig. 177, or the double-bucket, Pelton-water-wheel arrangement may be employed as shown in Fig. 178.

The section of each nozzle is a circle at its inlet end and a rectangle at its outlet end, a construction which allows the outlets of the nozzles to fit together. The nozzles extend all round the wheel, and thus a practically continuous stream of fluid can enter the whole circle of turbine buckets.

The author understands that tests have been made on a 1800 - H.P. Stumpf turbine, and that good results have been obtained. The Allgemeine Elektricitäts-Gesellschaft, of Berlin, have undertaken the manufacture of the Stumpf steam turbine.

In the **Seger Steam Turbine** two wheels are employed, each of which is of the parallel-flow type, after the nature of a De Laval turbine wheel. The steam issues from nozzles, and passes in series through the buckets of the two wheels. The wheels are separated by a perforated diaphragm. A nozzle and part of two wheels are shown in Fig. 179, the arrows indicating the direction of flow of the steam and the direction of rotation of the wheels which revolve in opposite directions.

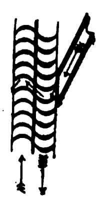


Fig. 179.—Showing Flow of Steam through Seger Steam Turbine.

Front and side elevations partly in section of a Seger turbine are shown in Figs. 180 and 181 respectively. The steam acts first on the wheel *b*, and then on the wheel *a*. These wheels are mounted on shafts *d* and *c*, which carry pulleys *f* and *e*. An endless belt passes round these pulleys and round larger pulleys *g* and *h*, of which the axes are perpendicular to the axes of the pulleys *e* and *f*. The wheel *a* rotates at about half the speed of the wheel *b*, and therefore the pulley *e* is made of about twice the diameter of the pulley *f*. The pulley *h* is mounted on the shaft *i*, which carries the pulley *k* (or which

may be coupled direct to the armature spindle of a dynamo). The tension of the belt can be regulated by raising or lowering the pulley *g*, which is provided only for this purpose and as a guide for the belt.

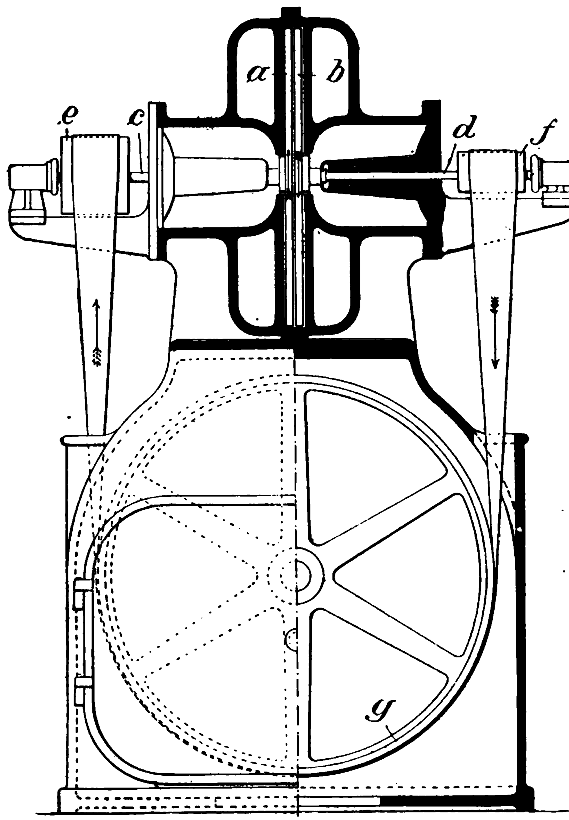


Fig. 180.—Seger Steam Turbine showing Belt-driving Arrangement.

Fig. 182 shows how a Seger turbine can be opened up for inspection. Fig. 183 shows a Seger turbine coupled to a dynamo. The hand wheels for controlling the flow of steam through the nozzles and the steam inlet and exhaust connections can be

seen in these figures. The illustrations of the Seger steam turbine are here reproduced by kind permission from *Le Génie Civil.*

Plate XXIII. shows an experimental **Schulz Steam Turbine.**

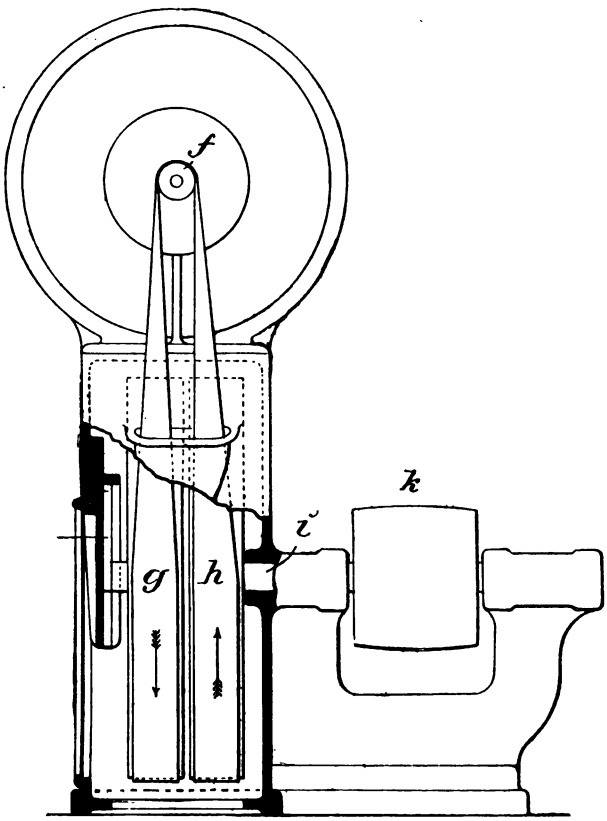


FIG. 181.—Seger Steam Turbine showing Belt-driving Arrangement.

This is a compound multiple-expansion parallel-flow turbine. The high-pressure turbine casing is seen at the right, and the low-pressure at the left. Steam passes alternately through guide-rings and through rings of moving blades. Slides are

provided which act to limit the passages in the guide-rings so as to control the flow of steam through the turbine according to the load. The lever by which these slides are actuated can be seen on the top and at the right-hand end of the high-pressure casing. A quadrant scale is provided to indicate the position of the slides.

Provision is also made for controlling the flow of steam

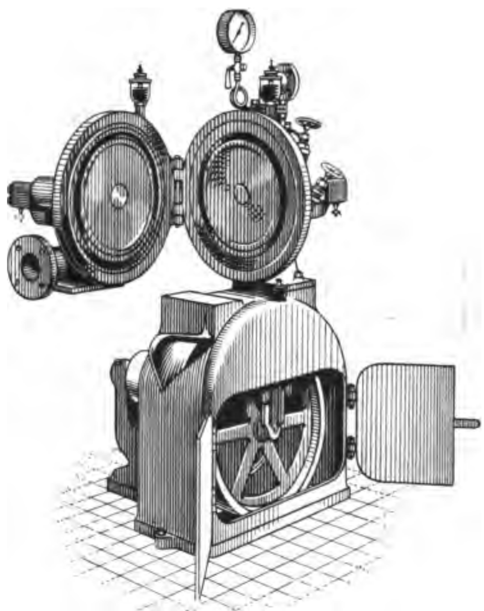


FIG. 182.—Seger Steam Turbine opened up for inspection.

through any one of the first ten guide-rings of the high-pressure casing independently of the others. This is done by means of the vertical screwed rods seen in the side of the high-pressure casing.

Steam-expansion diagrams for the high and low pressure turbines are shown in Figs. 184 and 185 respectively. These

diagrams were obtained by putting an indicator successively in communication with the several chambers of the turbine in which rotate the several rings of moving blades. The horizontal lines in the curve indicate the pressure at the corresponding rings of moving blades, and the vertical lines

FIG. 183.—Seger Steam Turbine coupled to Dynamo.

show the drop in pressure in passing from one chamber to the next beyond it.

The indicator is put in communication with the several chambers by means of an indicator cock and tubes, which can be seen in Plate XXIII. The cock with the tube connections is also shown in vertical section in Fig. 186, in front elevation

and section (without the handle) in Fig. 187, and (part) in back

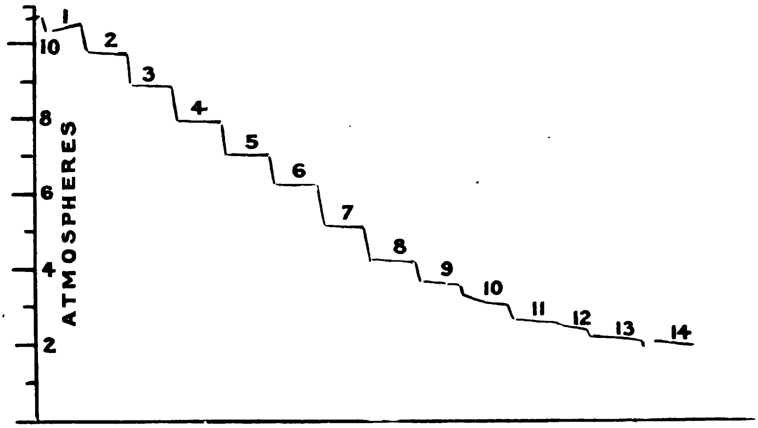


Fig. 184.—Steam-Expansion Diagram from Schulz Steam Turbine—High-pressure Cylinder.

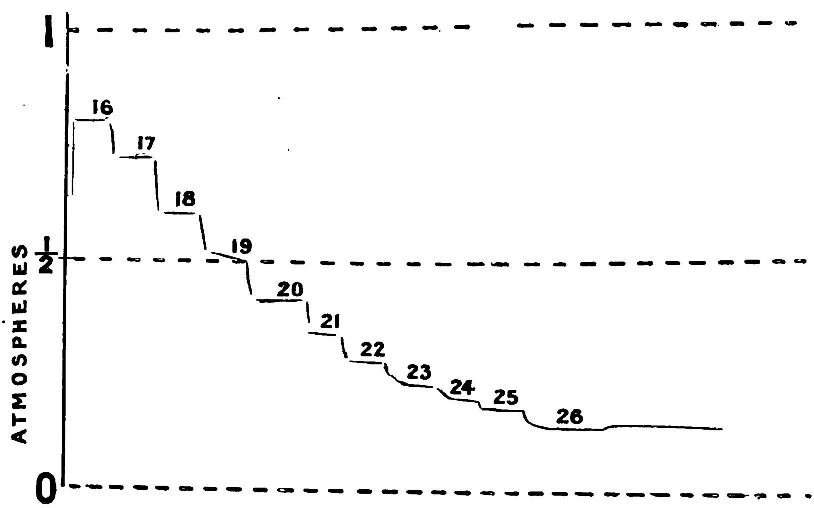


Fig. 185.—Steam-Expansion Diagram from Schulz Steam Turbine—Low-pressure Cylinder.

elevation in Fig. 188. The cord which rotates the indicator barrel is attached to and wound round the drum C, and the

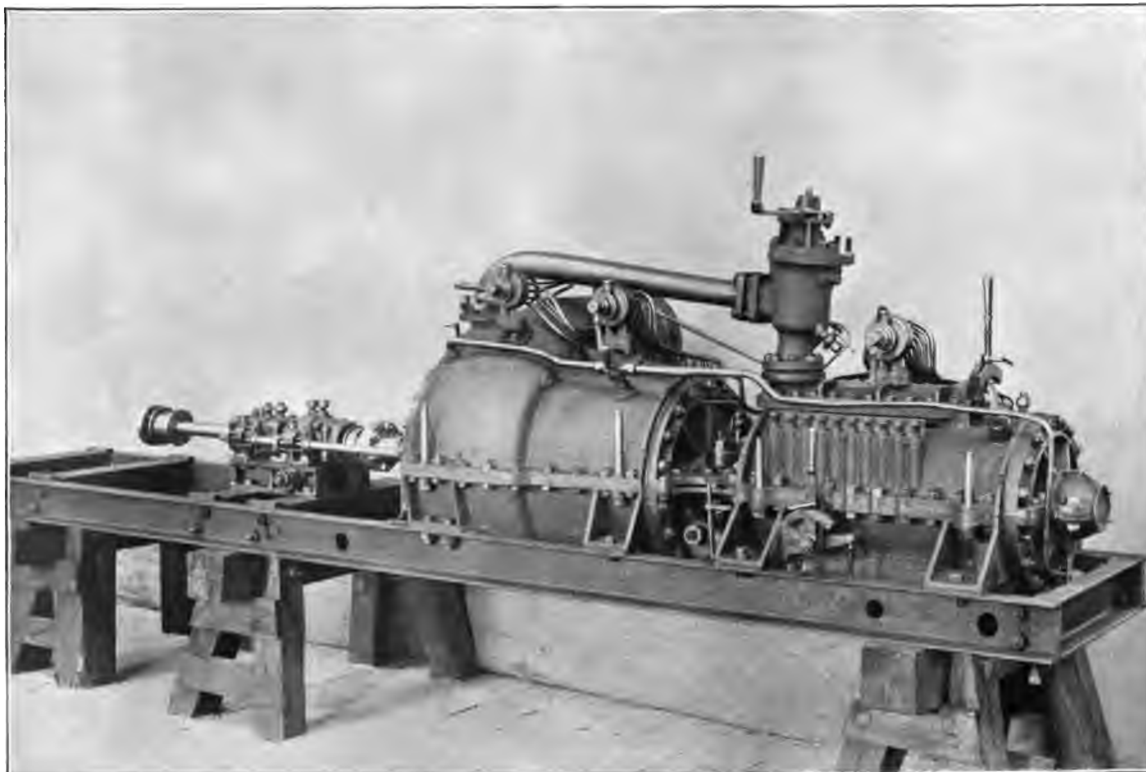


PLATE XXIII.—COMPOUND SCHULZ STEAM TURBINE.

handle D is then given a turn. The indicator nipple A is thus put successively in communication with all the tube nipples and the indicator barrel is rotated at the same time, thus producing the diagrams.

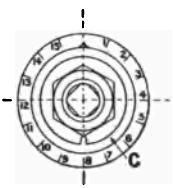
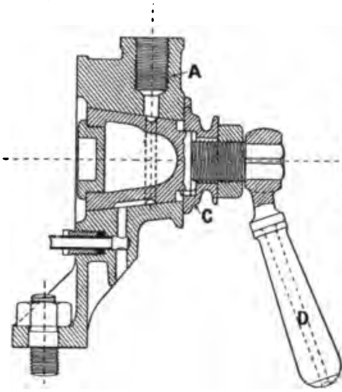


FIG. 188.



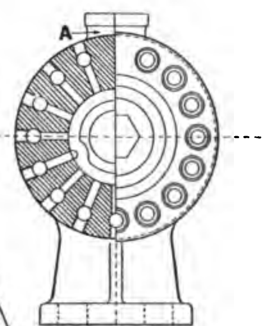


FIG. 186. FIG. 187.

Indicator Cock for obtaining Diagram of Steam Expansion in Turbine.

Figs. 189, 190, and 191 show the effect of restricting the passages through the guide-rings in different parts of the high-pressure turbine. In Fig. 189 the passages through the whole of the first ten guide-rings were restricted to one-sixth. In Fig. 190 this contraction of the passages was made at the first, fifth, and tenth guide-rings only, while in Fig. 191 the same contraction was made, but only at the first guide-ring. The effects are clearly seen in the figures.

With steam at about 200 lbs. abs., the consumption was found to be reduced about 20 per cent. by superheating to 320° to 350° C.

The Curtis Steam Turbine (originated by Mr. C. G. Curtis of New York, and developed by him and by the engineers of the General Electric Company of America at Schenectady) resembles the Parsons in so far that the steam passes alternately

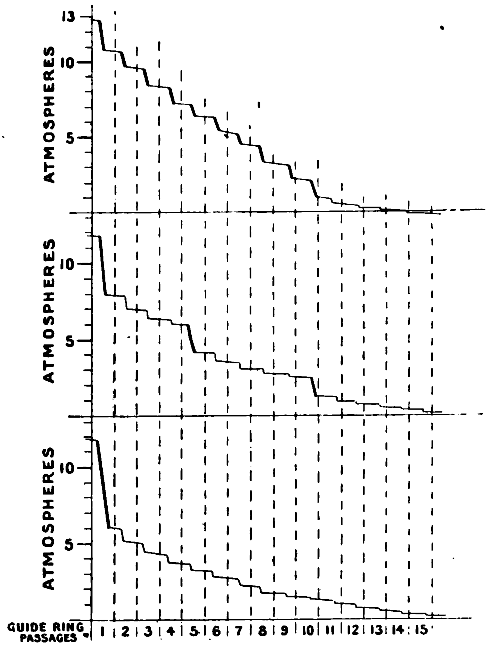


FIG. 189.

FIG. 190.

FIG. 191.

Diagrams showing Effect of restricting Passages through Guide-rings—Schulz Steam Turbine.

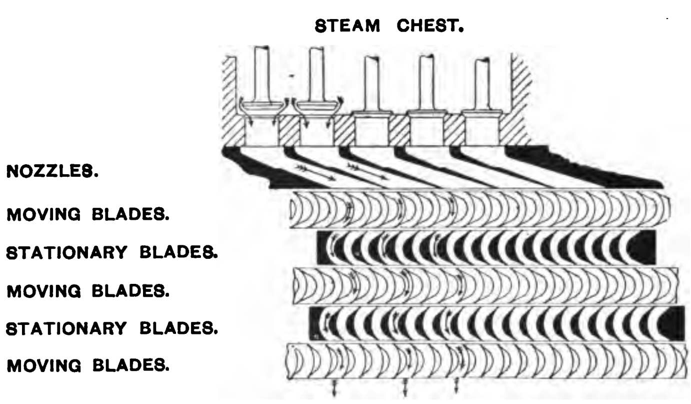


FIG. 192.—Nozzles and Blades of Curtis Steam Turbine.

through fixed and moving rings of blades. The method of conducting the steam to the first ring of moving blades is, however, different, the steam being admitted through inclined nozzles or passages and having a high velocity when it strikes the first ring of blades. Fig. 192 shows some of the nozzles and some of the rings of fixed and moving blades, the arrows indicating the path of the steam. The steam may be cut

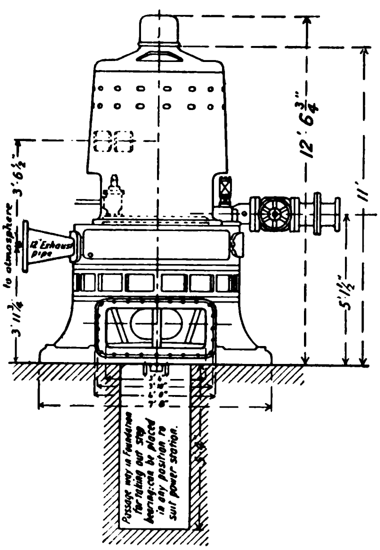


Fig. 193.—500-Kilowatt Curtis Steam Turbo-Generator. Front Elevation.

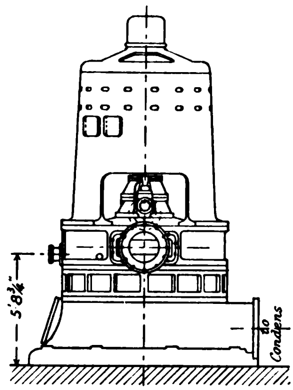


Fig. 194.—Side Elevation.

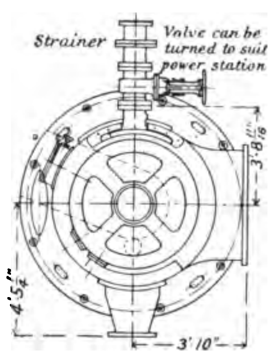


Fig. 195.—Plan.

off from some of the nozzles to reduce the power of the turbine.

The Curtis turbine as built by the General Electric Company of America for the driving of electric generators is of the parallel-flow type, and is arranged with a vertical axis of

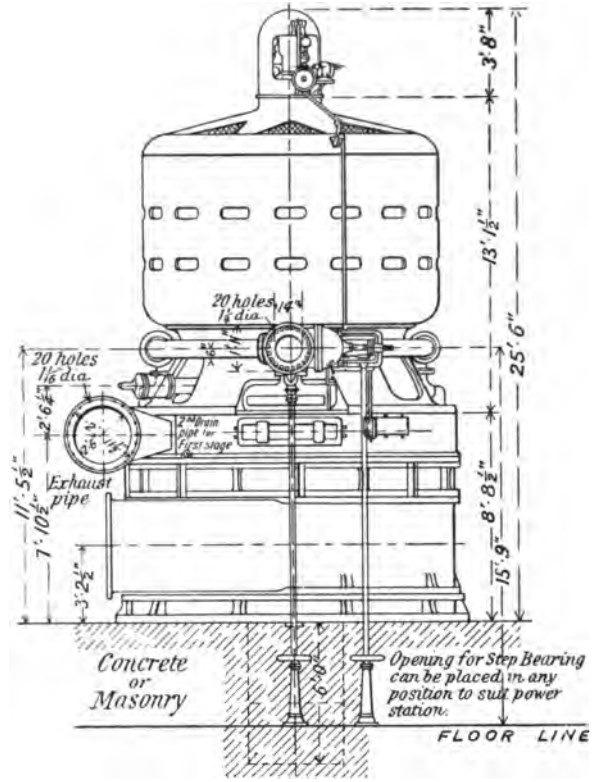


Fig. 196.—Front Elevation.

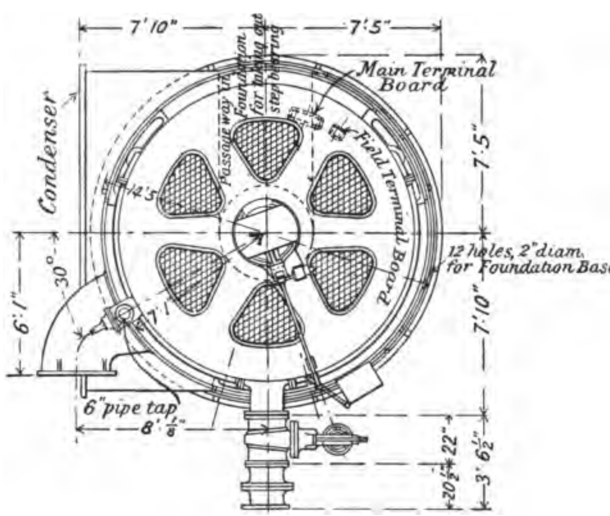


Fig. 197.—Plan.

5000-Kilowatt Curtis Steam Turbo-Generator.

rotation. The generator is placed over the turbine, and a very compact arrangement is obtained which requires an extremely small amount of floor-space for the power. A 5000-kilowatt Curtis turbo-generator at Chicago has a circular bed-plate only

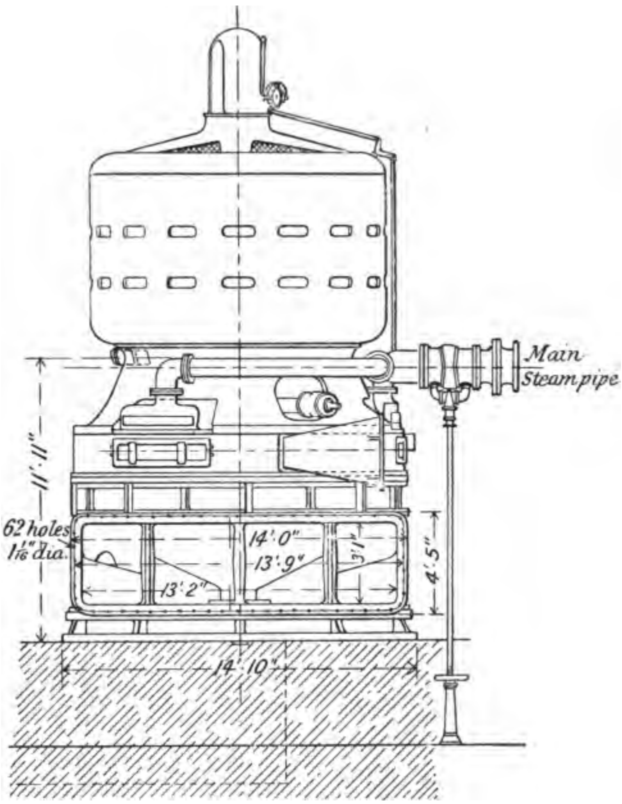


Fig. 198.—Side Elevation of 5000-Kilowatt Curtis Steam Turbo-Generator.

14 feet 10 inches in diameter. The extreme height of this machine, measured from the bottom of the bed-plate, is 25 feet 6 inches. Figs. 196, 197, and 198 show such a turbine.

Three Curtis steam turbines are installed in the Newport power station (Rhode Island) of the Massachusetts Electric

Companies. The turbines are of 1000 H.P., and drive 500-kilowatt three-phase alternators. The speed is 1800 revolutions per minute, and the voltage 2500. One of these machines is shown in Figs. 193, 194, and 195. The bed-plate is 7 feet 8 inches in diameter. The extreme height of each machine, measured from the bottom of the bed-plate, is 12 feet $6\frac{3}{4}$ inches. Steam enters each turbine at the top and leaves at the bottom, passing into a surface condenser, of which one is provided for each turbine. As the condensers are alongside the turbine foundations, and stairs are provided to lead to the generators, the saving in floor-space over that required by horizontal machines (such as Parsons) with the condensers arranged below them, is not so much as one might at first expect.

Tests made on the Curtis turbine seem to show that it gains in efficiency by the use of superheated steam and by a good vacuum to much the same extent as a Parsons turbine.

CHAPTER XIII.

SPEEDS AND SPACE.

THE high rotary speed of steam turbines is a disadvantage in many cases. As is pointed out in Chapter XIV., this high speed was a great obstacle to the application of the steam turbine to the driving of ships, although the difficulties have now been overcome. The high speed has also prevented the steam turbine being used for many purposes for which reciprocating engines are employed.

The high speed is, however, a distinct advantage in many cases. In the driving of rotary pumps and air propellers and compressors the high speed is usually advantageous. In the driving of electric generators, whether continuous current or alternating, high speeds mean (relatively speaking) small dimensions and small cost. When large-power Parsons turbines are used to drive continuous-current generators, the latter are made in duplicate to avoid commutator difficulties, the two dynamos being arranged tandem. This arrangement, although calling for more room, has its advantages.

In considering which of two motors to adopt for any purpose, the amount of space required by these is usually a consideration of some importance, and at times of very great importance. Table XXX. allows the floor-space required by steam turbines to be compared with that required by some vertical reciprocating steam-engines.

TABLE XXX.

FLOOR-SPACE OCCUPIED BY STEAM TURBINES WITH ELECTRIC GENERATORS, AND VERTICAL RECIPROCATING ENGINES WITH ELECTRIC GENERATORS.

Description of engine and generator.	Approximate floor-space of engine and generator without condenser and pumps.	Square feet of floor-space per kilowatt
Westinghouse Air Brake Co., Wilmerding, Pa. 300- K.W. Westinghouse-Parsons Turbo-alternator	75 square feet	0·25
Vertical Side-by-side Engine by D. Stewart & Co., Ltd. 16 in. and 32 in. by 30 in. driving 300-K.W. Generator	209 square feet	0·70
Curtis (vertical) Turbine driving 500-K.W. 3-phase Alternator at Newport, R.I., U.S.A.	46 square feet	0 09
Ferranti Vertical Engine and 750-K.W. Generator. Manchester Corp. Elec. Works	$23\frac{3}{4}$ feet × 15 feet	0·47*
Close Power Station, Newcastle-on-Tyne. One Parsons Turbine driving two Dynamos of a combined capacity of 1000 K.W.	232 square feet	0·23
Vertical Cross Compound Engine by D. Stewart & Co. Ltd. 35 in. and 71 in. by 42 in. driving 1500-K.W. Generator	650 square feet	0·43
Central Station of the Hartford Electric Light Company, Hartford, Conn. U.S.A. 1500-K.W.† Westinghouse-Parsons Turbine and Two-phase Alternator	291 square feet	0·19
Three-cylinder Compound Side-by-side Engine by D. Stewart & Co., Ltd. 38 in., 58 in., and 58 in. by 54 in. driving 2000-K.W. Generator	1230 square feet	0·61
Parsons Turbo-alternator for New Power Station of Sheffield Corporation. 2000 K.W.	37 feet × 9 feet	0·17*
Musgrave Vertical Engine driving 2500-K.W. Three-phase Alternator at Glasgow Corporation Tramway Power Station	58 feet × 24 feet	0·56*
Six-cylinder Tandem Engine by D. Stewart & Co., Ltd. Three $\frac{28}{59}$ in. by 54 in. driving 3000-K.W. Generator	1053 square feet	0·35
Parsons Turbine driving two Alternators having combined output of 4000 K.W.	400 square feet	0·10
Curtis (vertical) Turbine driving 5000-K.W. Generator at Chicago	173 square feet	0·035
5500-K.W. (rated ‡) Westinghouse-Parsons Turbo-alternator for the Metropolitan District Railway Company	724 square feet	0·13

* Taking floor-space area as length × breadth.

† Although rated at 1500 K.W., this machine gave on a four hours' test an average output of about 2000 K.W.

‡ Ultimate capacity expected to much exceed this.

The heights of two vertical reciprocating steam-engines are compared with those of two turbines in Table XXXI.

TABLE XXXI.

HEIGHTS OF STEAM TURBINES AND VERTICAL RECIPROCATING STEAM-ENGINES.

Type of engine.	Where situated.	Approximate B.H.P.	Extreme height in feet.	B.H.P. per foot of height.
Ferranti High-speed Vertical Reciprocating	Manchester Corporation Electricity Works	1200	15½	77·4
Musgrave Vertical Reciprocating	Glasgow Corporation Tramway Power Station	3700	34	108·8
Westinghouse-Parsons Steam Turbine	Wilmerding, U.S.A.	500	7¾	64·5
Parsons Steam Turbine	Milan	3000	9	333·3

The smallness of the head-room required by a steam turbine is, of course, of much less consequence when it is placed in an engine-room alongside of vertical reciprocating steam-engines than when located in an engine-room built to receive turbines only.

The floor-space and weights of steam turbines and gas-engines are compared in Table XXXII.

TABLE XXXII.

Steam Turbines compared with Large Gas-engines as regards Floor-space and Weight.

Nature of motor.	Type.	B.H.P.	Approx. over all length in feet.	Approx. over all breadth in feet.	Approx. weight, including flywheel, if any, in tons.	Weight per B.H.P. in pounds.
Gas-engine	Crossley	120	12	9	—	—
Steam Turbine	De Laval	150	11 Includes two broad-face driving pulleys	4⅓	4¾	70·9
Steam Turbine	De Laval	300	15 Includes two broad-face driving pulleys	6½	8	59·7
Gas-engine	Körting	400	30	18	50	280
Gas-engine	Gasmotoren-Fabrik Deutz	500	44	18¾	119	533
Gas-engine	Cockerill Single Cylinder	600	—	—	125	467
Gas-engine	Körting	700	41	23	127	406
Gas-engine	Körting	1000	49	29	190	426
Steam Turbine	Parsons Compound	3000	29½	8¾	35	26·1

CHAPTER XIV.

THE STEAM TURBINE APPLIED TO THE PROPULSION OF VESSELS.

THE success of the Parsons steam turbine on land led to the formation of a company in the beginning of 1894 for applying the steam turbine to marine purposes. This pioneer syndicate—the Marine Steam Turbine Co.—at once commenced experimental work, and the *Turbinia* was produced. It had often previously been proposed to use a steam turbine for the propulsion of vessels at sea; but, as far as the author is aware, no steam turbine was ever before fitted on board a vessel for this purpose. The same difficulty now arose with the marine steam turbine as had arisen with turbines previously made for use on land—namely, of running the turbine economically at a sufficiently low speed. In the driving of electric generators a high speed is usually an advantage, except when it becomes so excessive as to occasion dangerous stresses due to centrifugal force. With screw propellers, however, the case is very different. The existence of cavitation with high velocities of screw propellers was not unknown at the time the *Turbinia* was built; but the importance of it with propeller-blade velocities such as those tried in the *Turbinia* was not appreciated. The trials of the *Turbinia*, however, clearly demonstrated that an ordinary propeller could not be run with any degree of efficiency above a certain velocity. The propelling

gear of the *Turbinia* as first tried consisted of a single steam turbine driving a single propeller-shaft, on which were three propellers. The designed speed of the turbine was 3000 revolutions per minute, and the designed power was 2000 H.P. The power was obtained (as was proved by the use of a dynanometer); but, at the designed speed of rotation, only 18 knots could be got out of the vessel—the maximum efficient propeller velocity had been exceeded. Beyond this limiting velocity (the exact value of which depends on the size and form of the propeller) an almost perfect cylindrical vacuum is formed around the propeller, causing great loss of power.

As a steam turbine could not be run economically except at a high velocity—above the limiting velocity of a propeller—the difficulty arose of getting an efficient combination. With a low velocity the steam consumption was excessive; with a high velocity the waste of power by the propeller was enormous.

The designers of the *Turbinia* and her propelling gear, however, energetically and scientifically grappled with the difficulty. Trials were made with screws of various patterns, a spring torsional dynanometer was constructed and fitted between the turbine and the propeller-shaft to measure the actual torque, and a series of experiments were carried out in a tank with model propellers, which were illuminated by the light from an arc lamp thrown on to them for a single instant in each revolution. At length, after a great amount of labour, the efforts of the experimenters were crowned with success, a combination and arrangement of turbines and screw propellers being obtained which gave excellent results—results as good as the most optimistic of well-wishers had ever hoped for.

The solution of the difficulty was found in dividing up the power into three turbines driving three propeller-shafts. Each

PLATE XXIV.—THE PIONEER OF MARINE STEAM TURBINE PROPULSION.

shaft carried three propellers of a special form. As the economic speed of a turbine depends on the difference of pressure of the entering and exhausting steam, it will be obvious that, by dividing the total range of pressure into three parts—that is, in expanding the steam only about one-third in each turbine—the minimum economic speed of each machine could be very much reduced—in fact, reduced to about one-half. The propeller-shafts could thus rotate at one-half the speed. In addition to this, the employment of so large a number of propellers—nine in all—allowed each to be of small size, and therefore allowed the tips of the blades to revolve in circles of small diameters. By thus reducing both the size and the angular velocity of the propellers, and giving them a suitable design, their efficiency was brought quite up to the normal. The result was that the *Turbinia* attained a speed—33 to 34 knots—never before reached by any vessel.

The length of the *Turbinia* is 100 feet and the beam 9 feet. The displacement is 44½ tons, which is made up as follows:

Main engines, 3 tons 13 cwts.	
Total weight of machinery and boiler, screws and shafting, tanks, etc.	22 tons
Weight of hull complete	15 „
Coal and water	7½ „
Total displacement	44½ „

Steam is supplied by a water-tube boiler, and enters the first turbine cylinder at a pressure of 170 lbs. per square inch. The heating surface of the boiler is 1100 square feet, and the grate area 42 square feet. The stoke-holds are closed, and

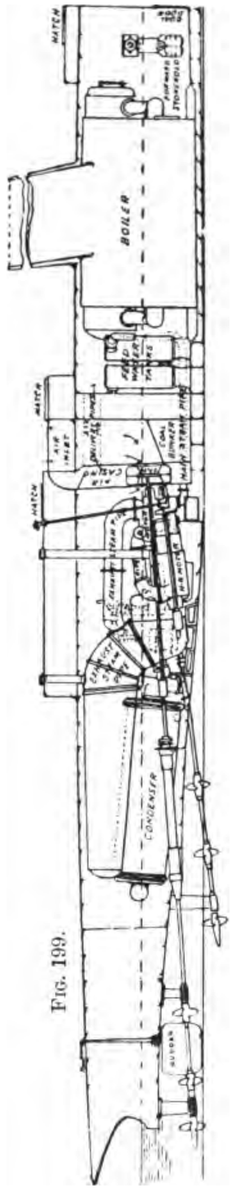


FIG. 199.

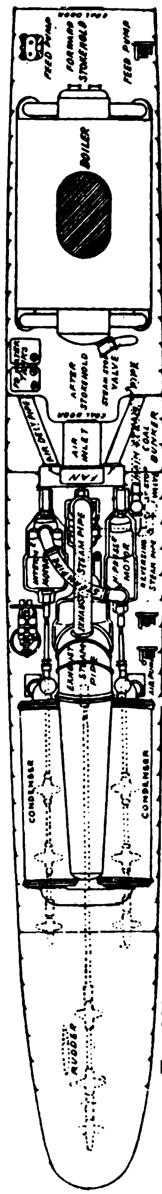


FIG. 200.

Arrangement of Machinery in the *Turbinia*.

draught is furnished by a fan coupled directly to the central shaft. 4200 square feet of cooling surface are provided in the condenser. The fresh-water tank and hot well contain about 250 gallons of water. The auxiliary machinery consists of main air-pump and spare air-pump, auxiliary circulating pump, main and spare feed-pumps, main and spare oil-pumps, and bilge ejectors.

The engine cylinders lie close to the bottom of the boat, and are bolted directly to small seatings on the frames. The reaction of the propellers and the axial thrust of the steam on the rotating parts of the turbine are arranged as far as possible to balance one another; but small thrust bearings are provided in the turbine bearings to withstand any difference or error of balance. Lignum-vitæ bearings are used for the propeller-shafts. The speed of rotation of the three shafts averages about 1200 revolutions per minute at 18 knots, about 2000 revolutions per minute at 30 knots, and about 2200 revolutions at 32 to 33 knots. Astern motion is given to the vessel by means of a reversing turbine situated on the central shaft. The arrangement of the machinery is shown in Figs. 199 and 200.

The propeller shafts are 2½ inches in diameter, and are inclined to the horizontal, the centre shaft having an inclination of about 1 in 16, and the others an inclination of about 1 in 8½. The propellers shown in the drawings are 18 inches in diameter and 24 inches in pitch. These are the ones referred to on page 245, and the speeds of rotation given in the preceding paragraph were obtained when these were in use.

In May, 1903, however, these nine propellers were removed, and a trial made with three propellers of 28 inches diameter and 28 inches pitch. These propellers are carried one on each

shaft beyond (that is aft of) the last bracket. The new arrangement proved a success, tests of speed and steam consumption showing a reduced consumption of water at the same speed, or an increased speed for the same consumption. The greatest advantage was found to be at about 20 to 25 knots. 23 knots with the new arrangement was obtained with about the same consumption of water per hour as 21 knots with the old arrangement.

The hull of the boat is built of mild steel plates, varying in thickness from $\frac{3}{16}$ inch at the bottom to $\frac{1}{16}$ inch at the sides near the stern. Water-tight bulkheads divide the vessel into five compartments.

The success of the *Turbinia*, which was only built for experimental and demonstrative purposes, led to the formation under the same directorate of a larger company—the Parsons Marine Steam Turbine Co., Ltd.—and the construction of the ill-fated torpedo-boat destroyers, *Viper* and *Cobra*. Of these the first was built to the order of the British Admiralty, who subsequently purchased the other after completion.

The *Viper* was 210 feet long, 21 feet beam, and 12 feet 9 inches moulded depth, the hull being constructed with the standard Admiralty scantlings for 30-knot destroyers, and further strengthened in parts for the higher speeds contemplated. The displacement was 350 tons. There were four shafts and two propellers on each shaft, the after propeller on each shaft having a slightly greater pitch than the forward one. On each side of the vessel a high-pressure turbine drove the outer and a low-pressure turbine the inner shaft. The inner shaft on each side was also fitted with a reversing turbine, the two reversing turbines being capable of driving the vessel astern at a speed of 15 knots. Plate XXV., reproduced by

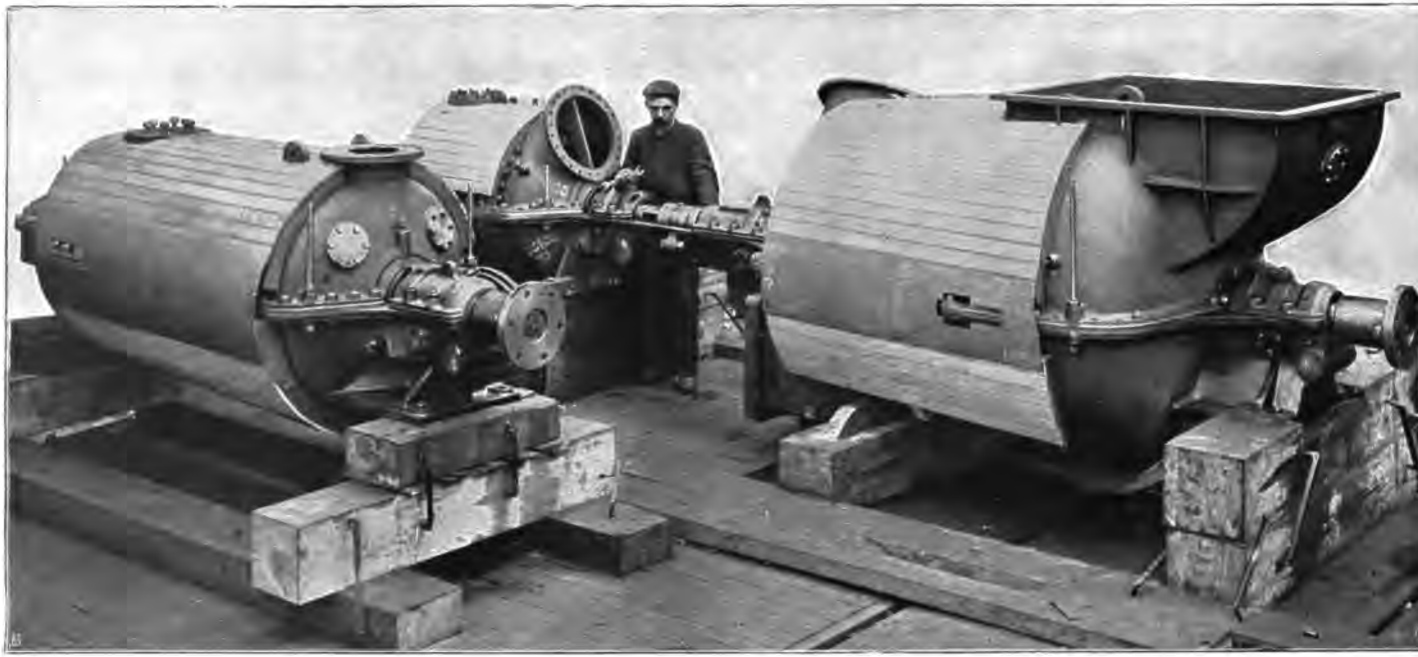


Plate XXV.—One Set of Engines for H.M. Torpedo-boat Destroyer "Viper" supplied by the Parsons Marine Steam Turbine Company, Limited.

(*From "Engineering," by kind permission.*)

kind permission from *Engineering*, shows one set of turbines. The cylinder on the left is the high-pressure turbine, and the one to the right on the other shaft is the low-pressure turbine, which receives the steam which exhausts from the high-pressure cylinder. The small cylinder at the back is the reversing turbine. The set of engines for the other side of the vessel was similar. Steam was supplied by four Yarrow boilers, having a total heating surface of 15,000 square feet and a total grate area of 275¾ square feet. The thrust of the propellers was arranged to balance the thrust of the turbines. The fittings were constructed to satisfy Admiralty requirements, and were much the same as those of other destroyers. The diameter of each high-pressure cylinder was 35 inches, and of each low-pressure cylinder 50 inches. The weights of boilers and machinery are as follows:—

Boiler-room weights with water in boilers...	120 tons
Engine-room weights with auxiliary gear and water in condensers	65 "
Propellers, shaftings, etc.	8 "
Total	193 "

Although the contract for the whole vessel was given by the Admiralty to the Parsons Marine Steam Turbine Co., Ltd., that firm, while themselves making and fitting on board the engines, sublet the contract for the hull and boilers to Messrs. Hawthorne, Leslie and Co.

On her official steam trials under the direction of the Admiralty officials, the *Viper* easily attained a speed of 33·838 knots on a three-hours' run. At this speed, the consumption

of coal was 11 tons 9 cwt. 1 qr. 9 lbs., or 25,685 lbs. per hour. On a three-hours' trial at 31·118 knots, the coal burned per hour was 19,846 lbs.

At a preliminary trial instituted by her contractors, the *Viper*, with a displacement of 380 tons, attained a mean speed on two runs with and against the tide of 36·849 knots. The mean speed for an hour's run alternately with and against the tide was 36·581 knots, the mean revolutions being 1180 per minute. The steam pressure during the six-hours' run ran up to 200 lbs., and the mean air-pressure in the stokeholds was 4½ inches. The speed was changed from 10 knots to 36·585 knots in twenty minutes.

The *Viper* was wrecked, it will be remembered, off Alderney in a fog, during the naval manœuvres in the summer of 1901.

The *Cobra* was built by Sir W. G. Armstrong, Whitworth and Co., Ltd., and engined by the Parsons Marine Steam Turbine Co., Ltd. This boat was slightly larger than the *Viper* (although of less beam). Her engines being similar in size and power, she was not quite so speedy. The length was 223 feet 6 inches; beam, 20 feet 6 inches; draught, 6 feet; displacement, 400 tons. The *Cobra* foundered during a gale on September 18, 1901, while being taken from the Tyne to Portsmouth Dockyard to undergo trials by the Admiralty. She had three propellers on each of her four shafts—twelve propellers in all.

The first merchant steamer to be propelled by steam turbines is the *King Edward*, which commenced running in July, 1901. This vessel was built by Messrs. William Denny and Bros., of Dumbarton, and is engined with Parsons' turbines.

The dimensions of the vessel are as follows: length, 250 feet; beam, 30 feet; moulded depth, 10 feet 6 inches to the main deck, and 17 feet 9 inches to the promenade deck. Steam

is supplied by a double-ended return-tube Scotch boiler of the usual marine type, having four furnaces at each end. There are three propeller-shafts, of which the two outer ones each carry two propellers 40 inches in diameter and about 9 feet apart on the shaft. The central shaft is provided with only one propeller, which is 57 inches in diameter. The stern of the vessel and the propellers are shown in Figs. 201 and 202. The draught of the vessel is about 6 feet. A high-pressure turbine is situated on the central shaft, in which turbine the

FIG. 201.—Under-water part of the Stern of the *King Edward*.

steam supplied at 150 lbs. is expanded about 5-fold, and then passes to two low-pressure turbines on the wing shafts, where it is expanded about 25-fold, the total expansion, therefore, being about 125-fold. The air-pumps are driven by worm gearing from the wing shafts. Reversing is done by two turbines situated in the exhaust ends of the casings of the main low-pressure turbines. Steam can be supplied direct to the low-pressure cylinders, and the high-pressure turbine and its shaft cut out of use in order to obtain greater manœuvring power for negotiating piers. The weight of the motors,

condensers with water in them, steam-pipes, auxiliaries connected with the propelling machinery, shafting, propellers, etc., is 66 tons, which is very much less for the power developed than the propelling machinery of reciprocating-engine, paddle-propelled passenger steamers of the same type.

The *King Edward* was employed for passenger traffic between Fairlie and Campbeltown in the summer of 1901,

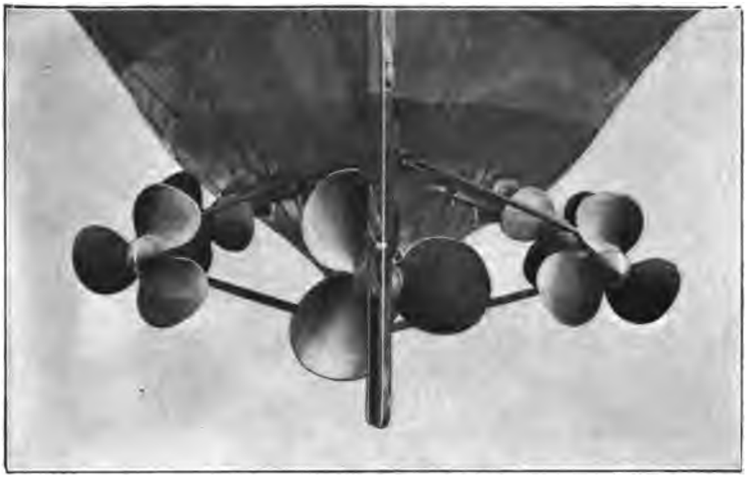


FIG. 202.—Under-water part of the Stern of the *King Edward*, looking forward.

and gave great satisfaction. The turbines produce no vibration whatever, a slight vibration aft being due to the propellers.

In the trials of the *King Edward*, on June 26, 1901, on the Clyde, a mean speed of 20·48 knots was obtained on several runs over the measured mile at Skelmorlie. The mean revolutions at this trial were 740 per minute. The steam-pressure at the boilers was 150 lbs., and the vacuum 26½ inches. The air-pressure in the stoke-hold was equal to 1½ inches of water.

The *King Edward*, after a very successful season, was transferred to another route (to Tarbert and Ardrishaig), and its

place taken by a larger turbine steamer—the *Queen Alexandra.* This vessel also was built by Messrs. Wm. Denny and Bros., and the engines supplied by the Parsons Marine Steam Turbine Co., Ltd. The length of the *Queen Alexandra* is 270 feet, and breadth (moulded) 32 feet. The depth to the main deck is 11 feet 6 inches, and to the promenade deck 18 feet 9 inches. Above the main deck there is a promenade deck which extends right to the bow and nearly to the stern, and above this again a shade deck which extends over 100 feet of the length of the vessel. The vessel draws about 6 feet 6 inches of water.

Steam is furnished by a large double-ended Scotch boiler supplied by Messrs. Denny and Co., the working pressure being 150 lbs. per square inch. The products of combustion pass away by two funnels, one at each end of the boiler. The steam is expanded about 5-fold in the high-pressure turbine arranged on a central shaft, carrying one propeller about 4 feet in diameter. The steam then divides, and proceeds in parallel through two low-pressure turbines, where it is expanded another 25-fold. The low-pressure turbines are arranged one on each side of the high-pressure cylinder, and each drives a shaft which originally carried two propellers about 3 feet in diameter. There were thus five propellers in all. At the ordinary steaming speed of the vessel, the central shaft made about 700 revolutions a minute and the side shafts about 1000 revolutions per minute.

Astern motion is given to the vessel by two reversing turbines each situated in the exhaust end of a low-pressure main turbine casing. The condensers are arranged outside the low-pressure and reversing turbines. The arrangement is shown in Plate XXVI., which is from a photograph taken of the engines of the *Emerald,* which are similarly arranged. As is the case

with the *King Edward*, steam can when desired be supplied direct to the low-pressure turbines for turning and manœuvring, and the central shaft put out of action. 21·63 knots was obtained from the *Queen Alexandra* on her trial trip.

In the spring of 1903 single propellers of greater diameter were substituted for the tandem propellers on the wing shafts. This alteration is said not only to have reduced the vibration at the stern of the vessel, but also to have increased the speed and reduced the coal consumption. The *Queen Alexandra* has, therefore, only three propellers now.

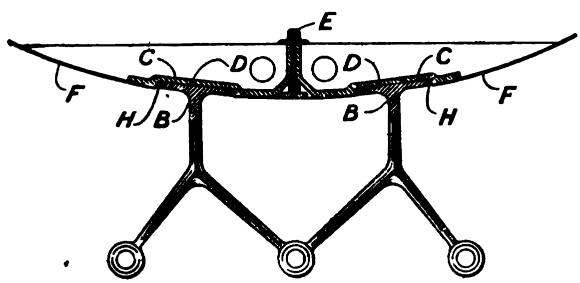


Fig. 203.—Propeller-shaft Support of Parsons and Wass: Sectional end elevation.

Figs. 203–206 illustrate a propeller-shaft support, recently patented by Messrs. Parsons and Wass, as applied to a vessel with a flat bottom upwardly inclined at the stern. Fig. 203 shows the support in end elevation, partly in section. Fig. 204 is a side elevation of part of the vessel with the support and propeller-shafts. Fig. 205 is a section on a line below the part of the vessel shown in Fig. 204. The support consists of two Y-shaped brackets of elliptical section, as shown at *a*, Fig. 205. The approaching arms of the two brackets are connected by a boss, while each of the outside arms also carries a boss. These bosses are lined with lignum-vitæ or white metal. Each bracket carries a sole, B, which is placed in a socket, C, in a

sole-plate, D, which is machined to receive it. The sole-plates are preferably formed of cast steel, and are permanently attached

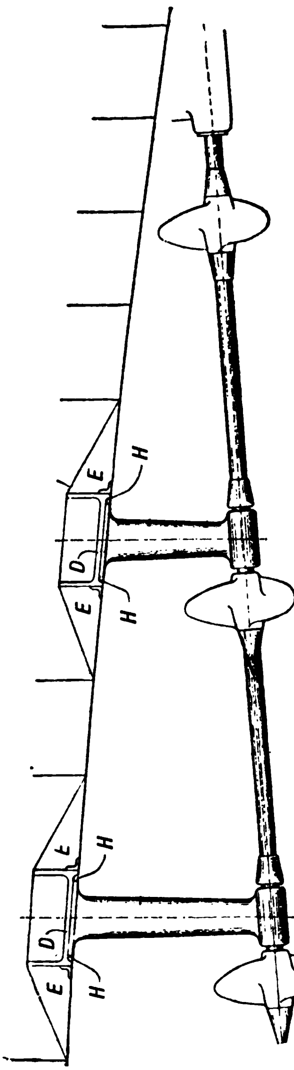


FIG. 204.—Propeller-shaft Support of Parsons and Wass: Side elevation.

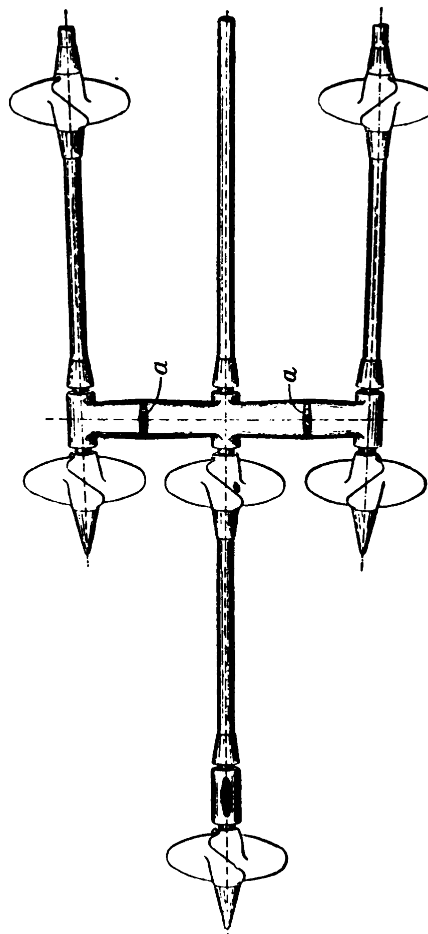


FIG. 205.—Propeller-shaft Support of Parsons and Wass: Sectional plan.

to the framing E and plates F of the vessel, the plates being cut away to allow of the insertion of the soles. If the brackets are

formed of aluminium bronze, manganese bronze, or gun-metal, strips H are provided round the soles to prevent corrosion. The end support for the central shaft is shown in Fig. 206. An

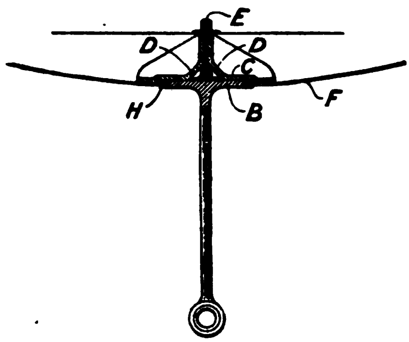


FIG. 206.—Propeller-shaft Support of Parsons and Wass: Rear support of centre shaft.

arrangement of brackets for four propeller-shafts is shown in Fig. 207.

It will be seen that these propeller-shaft supports will offer

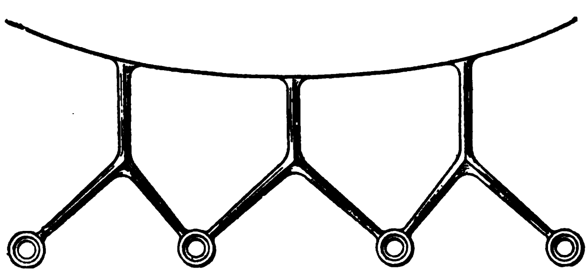


FIG. 207.—Support for Four Propeller Shafts.

very little resistance to passage through the water, and will be light and easily fitted correctly to the vessel.

Mr. Parsons states that he has found that the cavitation which attends high-speed propellers occurs principally in two places, namely, at the back faces of the blades near the tips,

and around the conical tip of the propeller-boss behind the blades. To obviate or lessen cavitation at the blade-tips,

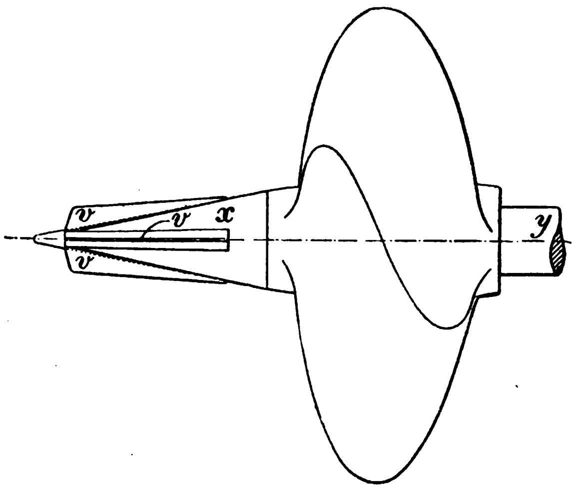


Fig. 208.—Parsons' Construction of Propeller Boss to Diminish Cavitation.

Mr. Parsons prefers to form the blades with diminishing pitch near the tips.

A device for diminishing cavitation round the conical end

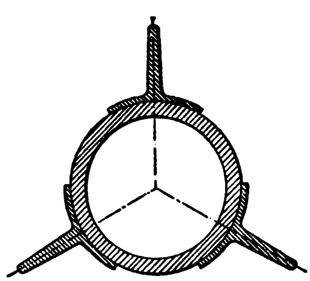


Fig. 209.—Cross-section of Boss.

of the boss has been patented by Mr. Parsons, and is shown applied to a propeller in Fig. 208. Vanes v are fixed on the conical end x, the vanes being parallel, or nearly so, to the axis

of the shaft *y*. Fig. 209 is a cross-section through the cone and vanes. The water put into rotation by the propeller-blades closes in on the cone *x*, but tends to retain its velocity. It therefore rotates with a greater angular velocity than the cone. The vanes *v* are, therefore, considered to produce two beneficial results. Firstly, some of the kinetic energy of the rotating water is given up to the shaft which it helps to rotate; and secondly, owing to the diminution of the velocity of the water rotating round the shaft, centrifugal force is reduced, and the water closes in more readily, and, pressing on the cone *x*, imparts an additional forward thrust to the shaft.

The steam yacht *Emerald*, built for Sir Christopher Furness, M.P., by Messrs. Alexander Stephen and Sons, Ltd., of Linthouse, to the designs of Mr. Fred. J. Stephen, is propelled by steam turbines supplied by the Parsons Marine Steam Turbine Co., Ltd. There are three shafts with one propeller on each The arrangement of machinery is much the same as that of the *King Edward* and *Queen Alexandra*. Plate XXVI. shows the actual engines. The small cylinder in the centre is the high-pressure turbine which drives the centre shaft. The two low-pressure turbines are arranged one on each side of the high-pressure cylinder, and reversing turbines are arranged inside the low-pressure casings. The condensers can be seen beyond the low-pressure cylinders on each side. For ordinary going ahead the steam passes first through the high-pressure turbine, and then in parallel through the low-pressure turbines, and thence to the condensers. When, however, the vessel is coming alongside a pier or is manœuvring, the high-pressure turbine is put out of action (and can rotate idly in a vacuum to prevent the drag of the centre shaft propeller), and steam is admitted direct to either or both of the low-pressure turbines.

Plate XXVI.—The Propelling Engines of the Steam Yacht "Emerald."

Plate XXVII. – The Turbine driven Yacht "Tarantula."

The steam can at will be admitted to either or both of the reversing turbines, and thus by rotating either of the outside shafts in either direction the vessel can be readily manœuvred. The starting platform is at the forward end of the engine-room, on a level with the turbines, and the controlling handles are grouped together so that they can be actuated from this platform. Entire control of the machinery is thus conveniently obtained.

The *Emerald* is 236 feet long over all, 28 feet 8 inches beam, and 18 feet 6 inches in depth. The tonnage is 756 tons yacht measurement. The hull and machinery were constructed under Lloyd's special 100 A1 survey. A range of deck-houses extends over more than half the length of the vessel, and on the top of these is a promenade deck which extends from side to side of the vessel. The boats are hung upon this deck.

The *Emerald*, after her official trials at Skelmorlie on April 10, 1903, made a successful voyage across the Atlantic, being the first steam turbine vessel to make such a journey. This voyage is important, as proving the suitability of steam turbine engines for moderate speeds and rough weather.

The steam turbine yacht *Tarantula* was built for the late Colonel McCalmont by Messrs. Yarrow and Co., Ltd., the naval architects being Messrs. Cox and King, of London. It is an exact model of a number of first-class torpedo-boats built by Messrs. Yarrow, except for the necessary alterations to suit the propelling machinery. The length of this yacht is 160 feet, and the beam 16 feet. Plate XXVII. shows the *Tarantula* previous to launching, and Figs. 210 and 211 show part of the stern of the vessel and the screw propellers. There are three propeller-shafts, and each shaft has three propellers. The centre shaft is driven by a high-pressure turbine which exhausts

into two low-pressure turbines which drive the side shafts. The steam turbines were supplied by the Parsons Marine Steam Turbine Co., Ltd. Steam is generated by two Yarrow small-tube boilers.

The large ocean-going steam yacht *Lorena*, of 1402 tons (yacht measurement), was built by Messrs. Ramage and Ferguson, Ltd., of Leith, from designs by Messrs. Cox and King, for Mr. A. L. Barber, of New York. The turbine engines were supplied by the Parsons Marine Steam Turbine Co., Ltd. The length of the vessel is given by the builders as 295 feet 4 inches over all, and 252 feet 4 inches on the water-line. The moulded breadth is 33 feet 3 inches, and the depth is 20 feet 4 inches. The draught is about 13 feet. A continuous promenade deck extends over nearly the whole length of the vessel. The propelling machinery is arranged very similarly to that of the four last-described turbine vessels. Each of the three shafts carries one propeller only. Steam is supplied by four cylindrical boilers, whose working pressure is 180 lbs. per square inch. The total heating surface is 8560 square feet, and the grate area 217 square feet. The boilers are fitted with Howden's system of forced draught. The vessel was designed for a speed of 16 knots.

Running several times over the measured mile at Aberlady, in the Firth of Forth, the *Lorena* attained a mean speed of just over 18 knots. The centre shaft made about 550 revolutions per minute, and the side shafts about 700. On this trial the machinery was kept running at full speed for about five hours, and worked with perfect smoothness. The yacht was in normal cruising sea-going trim, and had about 240 tons of coal on board.

The steam turbine, from its nature, can be run with best

Fig. 210.—The Stern of the *Tarantula*, showing the Propeller-shafts with Propellers and Supporting Brackets.

Fig. 211.—The Stern of the *Tarantula*, showing the Propellers and Propeller-shaft Brackets.

PLATE XXVIII.—STERN OF THE "TARANTULA."

efficiency only at one speed for the same initial and final pressures. If a vessel is intended to run normally at its maximum, the only difficulty in applying steam turbines to drive it is to get them to impart to the propeller shafts the requisite angular velocity. This difficulty was overcome when the *Turbinia* was first made a success, as has already been described.

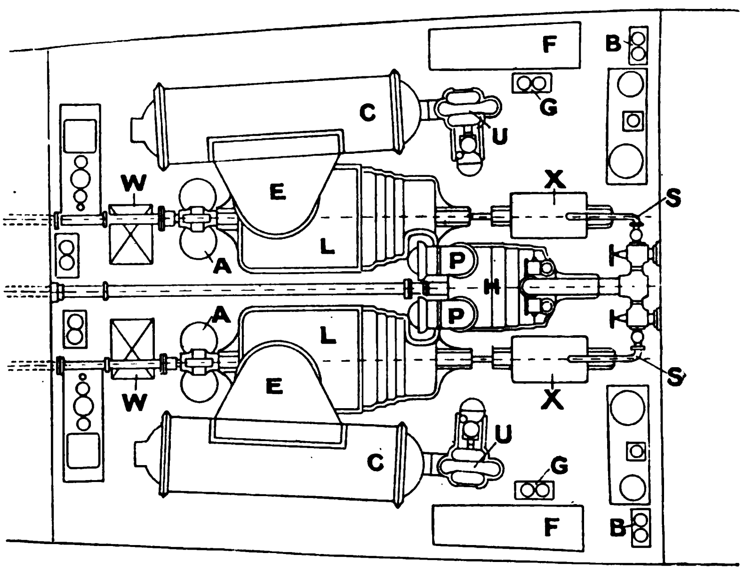


FIG. 212.—Arrangement of machinery on turbine-propelled vessel.

When, however, a vessel is intended to run normally or frequently at a speed considerably below its utmost speed, the difficulty arises of getting it efficiently propelled at both speeds by steam turbines. The propeller-shafts must obviously rotate at different velocities for the two speeds of the vessel (unless the pitch of the blades is altered, which would not only be mechanically difficult, but would be inefficient). Now, if the

turbines are designed to suit the high propeller speed, they will be less efficient at the low speed, and *vice versâ*.

This difficulty does not occur with passenger steamers, which normally run at about their full speed, but it does occur with war-vessels, which usually cruise at a comparatively low speed, and while employing only a small fraction of their total power. The difficulty has been overcome by employing auxiliary or cruising engines in addition to the main engines. In the third-class cruiser *Amethyst*, and in the destroyer *Eden*, these cruising engines consist of small steam turbines. In the destroyer *Velox* they consist of reciprocating engines. The former arrangement seems the better, and will probably be generally adopted in future turbine vessels requiring auxiliary cruising engines.

The destroyer *Velox* is 210 feet long, 21 feet wide, and 12 feet 6 inches moulded depth. The hull was built by Messrs. Hawthorn, Leslie and Co. to the orders of the Parsons Marine Steam Turbine Co., Ltd., who themselves supplied the engines, and from whom the vessel was bought after completion by the British Admiralty.

The *Velox* is provided with two small sets of triple-expansion engines in addition to her main turbine propelling engines. When the vessel is running at full speed, the reciprocating engines are not used, and the turbines rotate the propeller-shafts at an efficient turbine speed. When the vessel is cruising, the reciprocating engines are coupled up to two of the propeller-shafts, and the steam passes first through the reciprocating engines, then through the high-pressure turbines (of which there are two in this vessel), and then through the low-pressure turbines. As the steam is expanded in this case to a great extent before reaching the turbines, the range of steam-pressure

in the latter is comparatively small, and therefore the turbines can rotate efficiently at the low speed necessary for propelling the vessel at the cruising speed. The reciprocating engines are only required to be of small power, and, as the expansion of the steam is not completed in them, their bulk is comparatively small. The *Velox* has four propeller-shafts with two propellers on each. The high-pressure turbines drive the outer shafts, and the reversing turbines are arranged inside the casings of the low-pressure turbines which drive the inner shafts. Steam is supplied by Yarrow boilers, having a heating surface of 13,000 square feet.

The destroyer *Eden* was also built by Messrs. Hawthorn, Leslie and Co., and engined by the Parsons Marine Steam Turbine Co., Ltd. The *Eden*, however, unlike the *Velox*, was constructed to the order of the British Admiralty. The length is 220 feet, beam 23 feet 6 inches, and depth 14 feet 3 inches. There are three propeller-shafts, the centre one being driven by a high-pressure turbine, and each of the side ones by a low-pressure turbine. The reversing turbines are arranged inside the low-pressure turbine casings. The vessel is provided with auxiliary cruising steam turbines. Steam is supplied by Yarrow boilers.

The *Queen* is a turbine-driven vessel built by Messrs. Wm. Denny and Bros. for the cross-Channel service between Dover and Calais. The *Queen* is 310 feet long, 40 feet beam, and 25 feet in height to the promenade deck. The steam turbines were supplied by the Parsons Marine Steam Turbine Co., Ltd. On her trials, the vessel when running at 19 knots was brought to a dead stop in 1 minute 7 seconds, and in two and a half times her own length.

Fig. 212 shows in plan how the turbines are arranged in

several of the turbine-propelled vessels recently built. One high-pressure turbine, H, is arranged on the centre line of the vessel. This exhausts by the pipes P, P into the low-pressure turbines L, L. These exhaust by the large passages E, E into the condensers C, C. The air-pumps are shown at A, A, and the hot wells at W, W. F, F indicate the feed-tanks, G, G the hot-well pumps, B, B the bilge pumps, and U, U the circulating pumps.

X, X are the auxiliary cruising turbines which are of greatest use on war-ships, but may with advantage be placed on very high-speed steam yachts intended to cruise extensively at much reduced powers. (The *Tarantula* is provided with a cruising turbine to increase the economy at speeds up to 15 knots.) S, S are the steam supply pipes to these cruising turbines. When these auxiliary engines are omitted, the air-pumps are sometimes placed forward of the high-pressure turbine. The reversing turbines are placed inside the casings of the low-pressure turbines, and exhaust into the condensers by the same passages E, E. The pipes conducting live steam to the reversing turbines are omitted from the drawing for the sake of clearness.

The steam turbine possesses several advantages over the reciprocating engine for marine propulsion. In the first place, there is the absence of vibration—an important point, both as regards comfort in passenger steamers and as regards accuracy of gun-fire in naval vessels. Then there is a distinct saving in weight. This is not so marked in vessels of the destroyer type, where the engine-room weights are cut down to an abnormally small amount, as in larger vessels, and especially in the mercantile marine. This saving in weight can, of course, be used either in increasing the engine power, and consequently the

speed, of the vessel, or in adding to its carrying capacity. The low situation of the engine-room weights in a turbine-propelled vessel also tends to improve the stability, and, in the case of a war-vessel, places the engines in a more protected position. Turbines require less attention than reciprocating engines, and are easier kept clean and bright.

APPENDIX I

BRITISH PATENTS FOR OR RELATING TO STEAM TURBINES FROM THE EARLIEST RECORDS UP TO THE END OF 1903.

When inventions have been communicated from abroad, the names of the communicators are printed within parentheses.

1784.

1,426 . . Kempelen.
1,432 . . Watt.

1791.

1,812 . . Sadler.

1805.

2,887 . . Miller.

1809.

3,289 . . Noble.

1815.

3,922 . . Trevithick.

1823.

4,793 . . Peel.

1830.

5,910 . . Grisenthwaite.
5,961 . . Ericsson.

1831.

6,120 . . Hobday.

1834.

6,720 . . Craig.

1836.

7,242 . . Perkins.

1837.

7,305 . . Elkington.

1838.

7,554 . . Heath.
7,797 . . Burstall.
7,854 . . James.

1840.

8,474 . . Williams.
8,572 . . Cordes and Locke.

1841.

9,116 . . Jones.

1842.

9,354 . . Pilbrow.

1843.

9,658 . . Pilbrow.
9,902 . . Walther.

1844.

10,189 . . McIntosh.

1845.

10,765 . . Meade.

1846.

11,044 . . Taylor.
11,352 . . Bessemer.

1847.

11,800 . . Von Rathen.

1848.

12,026 . . Wilson.
12,080 . . Exall.
12,217 . . Stenson.

1850.

13,245 . . Barclay.
13,281 . . Fernihough.

1851.

13,598 . . Andrews.

1852.

14,351 . . Gorman.
149 . . Wheel.
776 . . Presson.
1,083 . . Slate.

1853.

480 . . Nicholls.
735 . . Brown.
2,768 . . Sochet.

1854.

315 . . Tourney.
944 . . Danchell.
1,706 . . Tetley.

1855.

2,747 . . Poulson.

1857.

2,076 . . Ivory.
2,598 . . Lombard.
3,061 . . Parker.

1858.

144 . . J. and E. Harthan.

1859.

805 . . Ivory.
1,041 . . Taylor.

1860.

119 . . Rutchet, Vonwiller, and Seiler.
1,155 . . Boyman.
2,317 . . Budden (Pilkington).

1861.

770 . . Chevillard.
2,457 . . Coffey.
2,953 . . Macintosh.

1862.

552 . . Parker.
1,568 . . Brakel, Hoehl, and Gunther.
3,252 . . Braddock.
3,283 . . Budden (Pilkington).

1863.

1,160 . . Thomson.
2,355 . . Lloyd.
2,692 . . Verran.

1864.

502 . . Southam.
2,596 . . Newton.
2,779 . . Galloway.

1865.

949 . . Brookes (Perrigault, Farcot, Farcot, Farcot, Château, and Farcot).
2,130 . . Stevenson (Venzano).

1866.

891 . . Wenner.
1,206 . . Newton (Farcot and Perrigault).
1,822 . . Fraser.
2,270 . . White (Sellier and Hermant).
3,289 . . Newton (Harris).

1867.

646 . . Clark, W. (Lemley, G. W.).
984 . . Moll, J. A.

1868.

784 . . Parker, J.
883 . . Beech, T. S. L.
1,732 . . Newton, W. E. (Boorman, J. M.).
2,320 . . Brooman, C. E. (Hielakker, J. V.).
2,680 . . Hunter, J. M.
3,146 . . Robertson, J.
3,307 . . Meldrum, R.
3,933 . . Lake, W. R. (de Amezaga, F.).

1869.

68 . . Legg, R.
208 . . Cook and Watson.
1,159 . . Brooman, C. E. (Goguel, E. F. A.).
1,748 . . Clark, A. M. (Lesnard, F.).
2,476 . . Mayall, J. J. E.
2,648 . . Muller. J. A.
2,830 . . Walker, W., and Davies, D.
3,267 . . Gorman, W.
3,642 . . Outram, J.
3,705 . . Bourne, J.

1870.

1,537 . . Astrop, W.
1,904 . . Lake, W. R. (Smith. J. Y.).
2,086 . . Scott, B. C.

1871.

1,736 . . Griffin, G. F.

1872.

2,188 . . Lake, W. R. (Harris, J.).
3,134 . . Robertson, J.
3,835 . . Cotter, R. H.

1873.

1,493 . . Burnett, W.
3,161 . . Baldwin, T.

1874.

706 . . Teulon, A.
3,961 . . Louche, J. H.

1875.

51 . . Turnock, J.
67 . . Boyman, R. B.
1,676 . . Newton, H. E. (Babbitt, B. T.).
1,848 . . Clark, A. M. (de Romilly, H. F. L. W.).
2,184 . . Preiswerk, L.
4,324 . . Preiswerk, L.

1876.

1,224 . . Pope, A.
1,549 . . Cotton, Sir A.
2,068 . . Edwards (Moorhouse).
2,368 . . Clark, A. M. (Dufort, J. H.).
3,483 . . Apperly, J.
3,841 . . Harris, J.

1877.

862 . . Apperly, J.
2,434 . . Lake, W.R. (Averseng, M. A. T.).
2,864 . . Smith, T. J. (Penning, G. A. de).

1878.

1,985 . . Brydges, E. A. (Bazin, R.).
4,293 . . Apperly, J.
4,596 . . Lumley, H. R.
4,628 . . Mills, B. J. B. (Gfeller, J.).
4,682 . . Tuckey, T.

1879.

409 . . Abel, C. D. (Binzer, J. von, and Bentzen, E.).
2,673 . . Davies, P.
3,521 . . Rigg, A.
5,022 . . Cutler, W. H.

1880.

17 . . Jensen, P. (Hahn, E. J.).
1,222 . . Prowett, W.
2,496 . . Howson, J. T., and Tate, W.
2,609 . . Nedden, F. zur.
3,522 . . Temple, G.
3,980 . . Jensen, P. (Hahn, E. J.).
4,160 . . Lake, W. R. (Cole, J. W.).

1881.

177 . . Imray, J.
255 . . Willet, T.
369 . . Temple, G.
981 . . Willet, T.
2,857 . . Leverkus, K. W. A.
5,237 . . Newton, H. E. (Desruelles, L. A. W., and Carlier, C. F.).

1882.

2,166 . . Charlton, G., and Wright, J.

1883.

911 . . Capell, G. M.
1,655 . . Engel, F. H. F. (Laval, G. de).
4,245 . . Johnson, J. H. (Delaurier, E. J.).
5,233 . . Lake, W. R. (Emmanuel, C.).

1884.

5,610 . . De Laval, G.
6,734 . . Parsons, Hon. C. A.
6,735 . . Parsons, Hon. C. A.
12,950 . . Dumoulin, A. J. A.

1885.

1,174 . . Johnson, J. H. (Howell, J. A., and Paine, F. H.).
3,885 . . Last, W. I.
4,483 . . Curtis, N. W.
8,773 . . Howson, J. T.

1886.

1,157 . . Neil, W.
5,647 . . Thévenet, J.
13,805 . . Tongue, J. G. (Brunner, A.).
13,949 . . Whittle, W.
16,020 . . De Laval, G.

1887.

5,312 . . Parsons, C. A.
9,591 . . Gwynne, J. E. A.
12,488 . . McConnell, J.

1888.

8,990 . . Thompson, W. P. (Erwin, J. B.).
9,158 . . Morton, A.
10,374 . . Kranich, F.
14,170 . . Hodgeman, H. D.
16,072 . . Haddan, R. (Dow, J. H., and Dow, H. H.).
17,299 . . Morton, A.

1889.

1,862	Curtis, N. W., and Carey, A. E.
4,302	Phillips, W. H.
5,619	Garside, A. A.
7,143	Laval, C. G. P. de.
8,884	West, J.
9,683	Howden, J., and Hunt, E.
9,684	Hunt, E.
12,509	De Laval.
13,593	Cousens, R. L. (Frost, W.).

1890.

291	Rowe, R.
1,120	Parsons, C. A.
2,050	Haddan, H. J. (Dow, J. H.).
2,691	Brown, J.W., and Sutcliffe, W. W.
5,768	Desgoffe, A., and Giorgio, L.
9,852	Sharples, P. M., and Sharples, D. T.
11,615	Moore, R. T.
14,994	Parsons, C. A.
15,264	Cot, J. P.
21,145	Allison, H. J. (Jones, J. H.).

1891.

4,596	Watkinson, W. H.
4,799	Thompson, W. P. (Altham, G. J.).
5,074	Parsons, C. A.
5,820	Morton, A.
10,940	Parsons, C. A.
20,449	Laval, C. G. P. de.
20,603	Laval, C. G. P. de.
21,376	Mossop, J.

1892.

10,370	Lake, H. H. (Altham, G. J.).
13,770	Laval, C. G. P. de.
15,677	Parsons, C. A.'
19,723	Justice, P.M. (Edwards, E. A., and Doughty, C. L.).
20,550	Rothery, G. W.
22,428	Scott, W. H.

1893.

2,720	Seger, E.
2,881	Nelson, W., and Niven, J. J.
7,807	Hutchinson, W. N.
8,357	Haddan, R. (Dow, J. H.).
8,854	Parsons, C. A.
15,703	Robinson, M. H.
17,297	Thompson, J. E., and Navard, E. J.
20,148	Beaumont, W. W.
22,573	Smith, I.
25,086	Raworth, J. S.
25,090	Raworth, J. S.

1894.

84	Raworth, J. S.
367	Parsons, C. A.
394	Parsons, C. A.
1,242	Raworth, J. S.
4,611	Seger, E.
6,248	Wrench, W. G.
6,822	Bollmann, L.
9,759	Haddan, R. (Piguet and Co.).
10,458	House, H. A., House, H. A., Symon, R. R.
11,526	Redfern, C. F. (Nordenfelt, P., and Christophe, A.).
11,880	Hopkins, G. M.

17,273 . . Lake, W. R. (Consolidated Car Heating Co.).
18,130 . . Larr, A. F. S. van de.
18,745 . . Rateau, A. C. E.
18,807 . . Vojacek, L.

1895.

2,565 . . Ferranti, S. Z. de.
3,506 . . Raworth, J. S.
11,709 . . Hewitt, J. T.
16,476 . . Grauel, H.
19,978 . . Jönsson, J. L.

1896.

24 . . Buchmüller, C.
180 . . Bollmann, L., and Kohnberger, S.
2,680 . . Benze, L., and Bachmayr, E.
6,073 . . Cook, D.
6,419 . . Capel, H. C., and Clarkson, T.
7,250 . . Bousfield, J. E. (Soc. des Provedes Desgoffe et de Georges).
7,455 . . Hewson, R., Whyte, N. C., and Rome, L. de.
8,697 . . Parsons, C. A.
8,698 . . Parsons, C. A.
8,832 . . House, H. A., and Symon, R. R.
11,086 . . Parsons, C. A.
11,351 . . Hayward, W.
12,060 . . Lacavalerie, S.
12,589 . . McAllister, J.
15,502 . . Davidson, S. C.
15,832 . . Dugard, W. H.
16,079 . . Dominy, G., and Sturmey, J. H.
17,136 . . Trossin, O.
17,481 . . Schmidt, J.
18,377 . . Ramstedt, C. W.
19,246 . . Mills, C. K. (Curtis, C. G.).
19,247 . . Mills, C. K. (Curtis, C. G.).
19,248 . . Mills, C. K. (Curtis, C. G.).
20,514 . . Jensen (Aktiebolaget de Lavals Angturbin).
22,369 . . Mackintosh, J.
26,612 . . Hug, D.
28,196 . . Fischer, A., and Held, A.

1897.

901 . . Parsons, C. A.
2,123 . . Martindale, M. D.
2,595 . . Ringelmann, M.
2,817 . . Weichelt, C.
6,800 . . Martin, H. M.
6,831 . . Heys, W. G. (Cazin, F. M.).
7,979 . . Martindale, M. D.
9,340 . . Stone, J. H.
10,284 . . Philipp, O.
10,609 . . Fiedler, L. R.
11,223 . . Parsons, C. A.
11,328 . . Hickson, E. (Hickson, S. A. E.).
12,529 . . Johnson, J. Y. (Sharples, P. M.).
14,885 . . McAllister, J.
15,069 . . Hakansson, L. M.
15,983 . . Ulenhuth, E.
16,635 . . Lohmann, C. F. C.
17,842 . . Marconnet, G. A.
19,673 . . Hayot, L.
20,536 . . Mills, C. K. (Curtis, C. G.).
22,226 . . Seger, E.
22,431 . . Senior, T. E.
22,842 . . Seger, E.
23,832 . . Huggins, W., and M'Callum, D.
24,113 . . Grubinski, F. von.
26,553 . . Parsons, C. A.
26,650 . . Jourdanet, A., and Gauthier, J. P.

26,669	. .	Gray, T. M., and Bass, F.
28,812	. .	Boyd, F. A.
28,821	. .	Thompson, W. P. (Irgens, P., and Brunn, G. M.).
29,508	. .	Huber, C.
29,637	. .	Scott, J.

1898.

3,068	. .	Miles, R.
3,455	. .	Clarke, W. H., and Warburton, F. J.
4,102	. .	Stuart, H. A.
4,714	. .	Addington, A. M.
4,922	. .	Thorssin, J.
4,932	. .	Stone, J. H.
7,398	. .	Stolze, F.
7,580	. .	Groterjam, C.
8,588	. .	Stone, J. H.
9,024	. .	Clarke, W. H., and Warburton, F. J.
9,044	. .	Paige, J. W., and Dixon, T. S. E.
9,220	. .	Yates, J., and Bellis, T. K.
10,503	. .	Schulz, R.
11,055	. .	Schulz, R.
11,159	. .	Canning, A. H.
11,668	. .	Petersson, F. O., and Franc, C.
17,271	. .	Johnson, C. M.
19,025	. .	Thompson, W. P. (Prall, W. E.).
19,256	. .	Bök, N. S.
19,350	. .	Montag, G., Hüter, F., and Karb, M.
19,392	. .	Bäckström, C. A.
19,394	. .	Lohmann, C. F. C.
20,099	. .	McCollum, J. H. K.
21,079	. .	Vandel, X. C. L. G.
21,478	. .	Davidson, S. C.
21,698	. .	Heys, W. G. (Heilmann, J. J.).
21,836	. .	House, I. M., and Overend, W. J.
24,084	. .	Prall, W. E.
24,204	. .	Pitt, S. (Rateau, A. C.E., and Sautter, Harlé, and Co.).
24,845	. .	Coard, J. B. M. A., and Charpentier, E. A.
26,721	. .	Bailly, P.
26,767	. .	Thrupp, E. C.
26,801	. .	Edge, H. T.

1899.

195	. .	Schroetter, J. F.
1,031	. .	Weihe, C. L.
1,149	. .	Gommerat, J. F., and Gommerat, L.
3,138	. .	Niepmann, F.
4,242	. .	Vijgh, G. van der.
4,638	. .	Enoch, A. G., and Enoch, D.
5,881	. .	Parsons, C. A.
6,768	. .	Baker, R. E., Dixon, T. H., Coghlan, J. B., Foley, E., Coleman, T., Dennehy, P. R., O'Brien, J., Crotty, J., Russell, E. B., Noonan, J., Mourissey, W., and O'Connell, M.
7,183	. .	Thompson, W. P. (Brady, J. F.).
9,119	. .	Jackson, J.
9,629	. .	Betscher, G.
10,296	. .	Lount, S.
10,980	. .	Billardon, A. L.
11,179	. .	Burgum, J.
11,433	. .	Haddan, R. (Rahmer, P.).
11,557	. .	Weichelt, C.
11,563	. .	Bruder, P.
14,476	. .	Parsons, C. A.
14,915	. .	Parsons, C. A., and Carnegie, A. Q.
15,724	. .	Spence, J.

15,954	Richards, R. S.
16,284	Parsons, C. A., Stoney, G. G., and Fullagar, H. F.
17,721	Nivert, E.
17,826	Paine, H. D., and Paine, E. G.
18,979	Zoelly, H.
19,839	Ferretti, E.
21,341	Thompson, W. P. (Brady, J. F.).
22,634	Taylor, C. H.
23,759	Nilsson, N.

1900.

2,400	Scott, J.
2,815	Whitcher, J.
4,295	Ashton, H. T.
5,198	Ashton, H. T.
7,116	Marburg, F.
8,295	Zoelly, H.
8,520	Ashton, H. T.
9,548	Ashton, H. T.
12,903	Thompson (Brady, J. F.).
14,038	Smiles, J. H.
15,130	Gravier, A. E.
16,551	Parsons, C. A.
16,603	Windhausen, F.
17,182	Riegel, A.
17,919	Newton, P. A. (Phœnix Investment Co., U.S.A.).
18,420	Kemble, D.
19,845	Jewson, H.
20,853	Nadrowski, J.
21,472	Schulz, R.
22,677	Graydon, J. W. and Greig, L. H.

1901.

2,096	Othon, L.
2,925	Hoffbauer, F., and Rüdemann, L.
3,565	Gelder, M. van.
6,239	Robinson, C. T., and Prescott, S. J.
6,347	Schulz, R.
6,422	Thrupp, E. C.
6,469	Probst, J.
7,065	Parsons, C. A.
7,066	Parsons, C. A.
7,184	Fullagar, H. F.
8,378	Schulz, R.
8,440	Krank, A.
8,738	Sayer, R. C.
8,850	Lennox, A. B.
8,934	Fullagar, H. F.
11,701	Sautter, G. E., Harlé, H. A. E., and Rateau, A. C. E.
11,943	Correll, W.
12,347	Parsons, C. A.
13,428	Davies, R.
13,714	Bowring, H. E.
14,153	Stumpf, J.
14,154	Stumpf, J.
14,155	Stumpf, J.
14,326	Stumpf, J.
14,593	Fullagar, H. F.
14,594	Fullagar, H. F.
14,664	Schulz, R.
15,569	McIntyre, D.
16,025	Hörenz, F. O.
16,232	Webster, W. L.
16,523	Knorring, C. von, and Nadrowski, J.
17,098	Graydon, J. W.
17,199	Weichelt, O.
17,941	Soliani, N.
17,951	Stumpf, J.
18,402	Wilson, L.
19,568	Astor, J. J.
20,669	Dürr, F.
21,164	Cassel, E. F.
22,462	Skorzewski, W. von.
24,201	Masters, T. J.
25,135	Clarke, M., and Warburton, F. J.
25,144	Hoffbauer, F.
25,411	Stumpf, J.
25,413	Stumpf, J.
25,414	Stumpf, J.

1902.

121 . . Fidler, E.
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APPENDIX II.

EQUIVALENT MEASUREMENTS.

Abbreviations.

cm. signifies	centimetre.
mm. ,,	millimetre.
$cm.^2$,,	square centimetre.
sq. in. ,,	square inch.
kg. ,,	kilogramme.
lb. ,,	pound.
K.W. ,,	kilowatt.
E.H.P. ,,	electrical horse-power.
I.H.P. ,,	indicated horse-power.
F. ,,	Fahrenheit.
C. ,,	Centigrade (Celsius).
B.T.U. ,,	British thermal unit.
$\doteqdot$,,	approximate equality.

HEAT.

1 calorie = 3·968 B.T.U.

1 B.T.U. = 772 to 780 foot-lbs.

t° C. $= (\frac{9}{5}t + 32)^\circ$ F.

t° F. $= \frac{5}{9}(t - 32)^\circ$ C.

Absolute zero can be taken as 273° C. below zero C., 492° F. below freezing-point, and 460° F. below zero F.

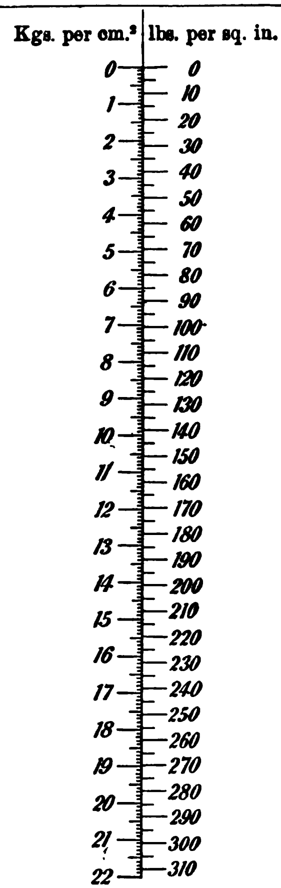
STEAM PRESSURE.

1 kg. = 2·2046 lbs. ≑ $2\frac{1}{5}$ lbs.
1 lb. = 0·4536 kgs. ≑ $\frac{5}{11}$ kgs.
1 sq. in. = 6·4516 cm.² ≑ $6\frac{1}{2}$ cm.²
1 cm.² = 0·155 sq. in. ≑ $\frac{2}{13}$ sq. in.
1 kg. per cm.² = 14·22 lbs. per sq. in.
1 lb. per sq. in. = 0·0703 kgs. per cm.²

TABLE A.

KILOGRAMMES PER SQUARE CENTIMETRE COMPARED WITH POUNDS PER SQUARE INCH.

(This table applies whether the pressure is absolute or reckoned above atmosphere.)

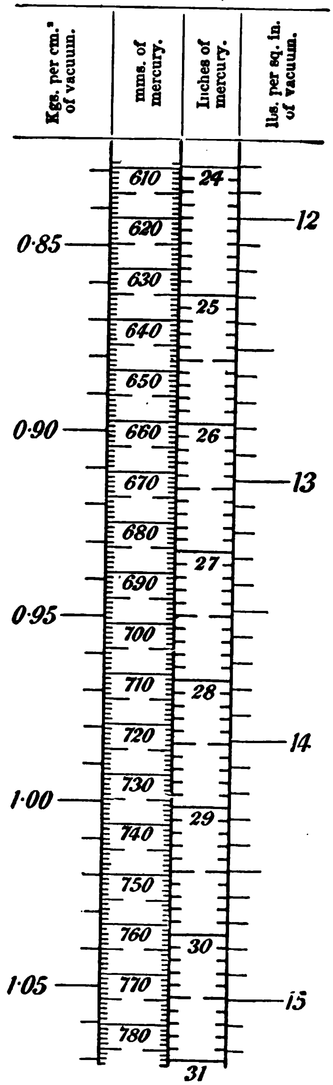


VACUUM.

TABLE B.

Comparison of Different Methods of expressing Degree of Vacuum.

(This table applies whatever the barometer is at)



1 inch of mercury = 25·4 mm. of mercury.
1 mm. " = 0·03937 inch of mercury.
1 lb. per sq. in. = 2·035 ins. of mercury.
1 inch of mercury = 0·491 lb. per sq. in.
1 kg. per cm.² (one metric atmosphere) = 735·5 mm. of mercury.
= 28·96 ins. "
= 14·22 lbs. per sq. in.

STEAM CONSUMPTION.

1 K.W. = 1·34 E.H.P. $\doteqdot$ $1\frac{1}{3}$ E.H.P.
1 E.H.P. = 0·746 K.W. $\doteqdot$ $\frac{3}{4}$ K.W.

TABLE O.

POUNDS OF STEAM PER HOUR PER KILOWATT COMPARED WITH THE SAME PER ELECTRICAL HORSE-POWER AND THE SAME PER INDICATED HORSE-POWER.

Per K.W.	Per E.H.P. (of 746 watts).	Per I.H.P. if $\frac{\text{E.H.P.}}{\text{I.H.P.}} = 0·80$	Per I.H.P. if $\frac{\text{E.H.P.}}{\text{I.H.P.}} = 0·85$	Per I.H.P. if $\frac{\text{E.H.P}}{\text{I.H.P.}} = 0·90$
14	10·44	8·36	8·88	9·40
15	11·19	8·95	9·51	10·07
16	11·94	9·55	10·15	10·74
16·5	12·31	9·85	10·46	11·08
17	12·68	10·15	10·78	11·41
17·5	13·05	10·44	11·10	11·75
18	13·43	10·74	11·41	12·09
18·5	13·80	11·04	11·73	12·42
19	14·17	11·34	12·05	12·76
19·5	14·55	11·64	12·36	13·09
20	14·92	11·94	12·68	13·43
21	15·67	12·53	13·32	14·10
22	16·41	13·13	13·95	14·77
23	17·16	13·73	14·58	15·44
24	17·90	14·32	15·22	16·11
25	18·65	14·92	15·85	16·78

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